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HIGH POWER FAST START TURBINE POWER UNIT.(U)

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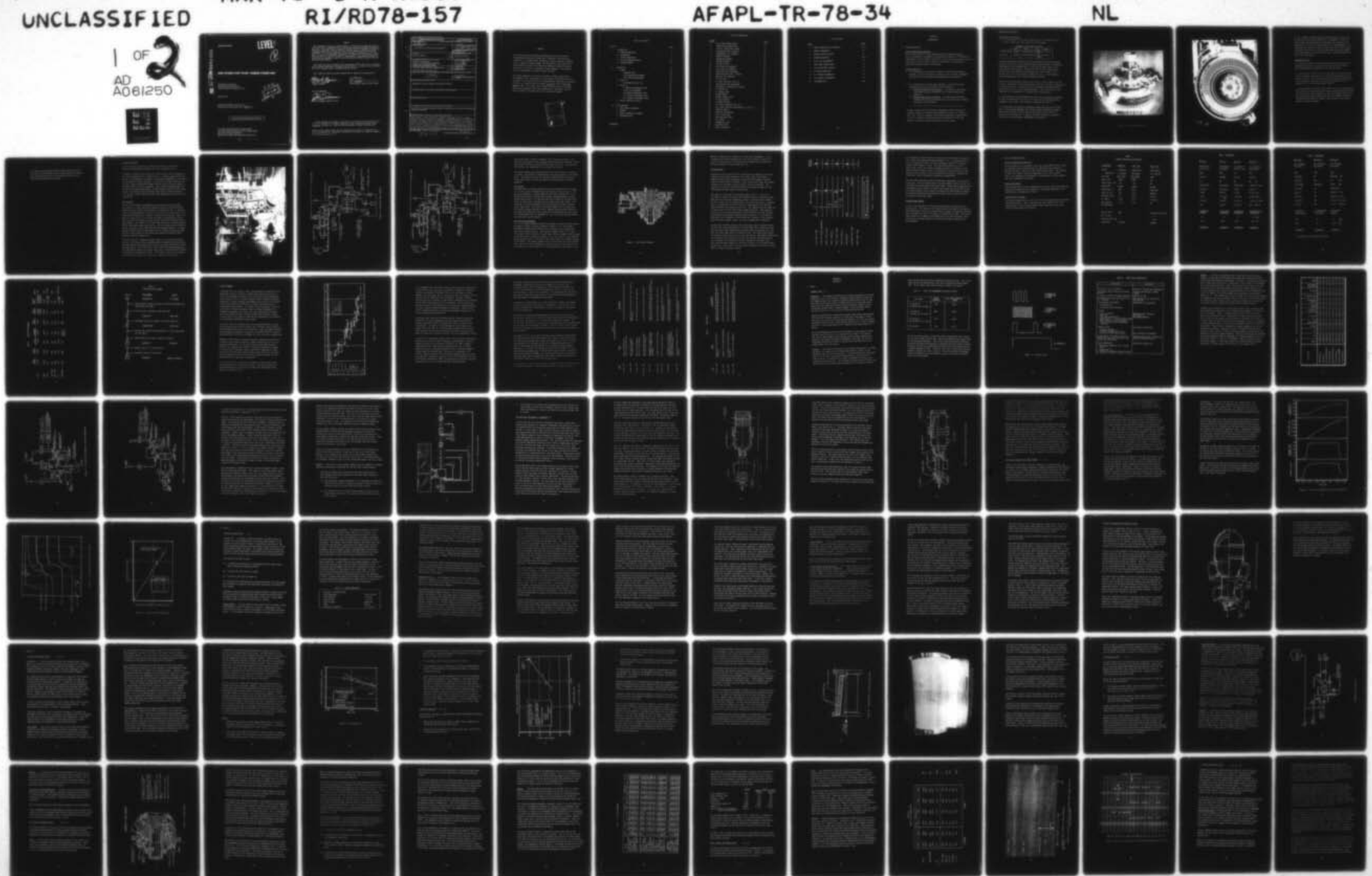


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AFAPL-TR-78-34

HIGH POWER FAST START TURBINE POWER UNIT

ROCKETDYNE DIVISION
ROCKWELL INTERNATIONAL
CANOGA PARK, CALIFORNIA 91304

MARCH 1978

TECHNICAL REPORT AFAPL-TR-78-34
Final Report for Period January 1974 – March 1978

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AIR FORCE AERO PROPULSION LABORATORY
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A three phase, 50 month program was conducted from January 1974 thru February 1978 during which efficient (5.04 lbs/hp-hr), lightweight (250 lbs) turbines were designed, fabricated, developed, acceptance tested and delivered. The resulting turbines were demonstrated to deliver 6000 shaft horsepower within 0.85 seconds when accelerated from rest to 29,000 RPM under a 0.225 slug-ft. parasitic inertial load. The driving power was provided by a fast starting hydrazine monopropellant gas generator, also developed under this contract. The results obtained here are directly applicable to the development of much larger gas generator and turbines.

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PREFACE

This final report was prepared by Rocketdyne Division of Rockwell International Corporation, Canoga Park, California. The report was written by D. R. Hodson and W. Studhalter, Project Engineers, approved by R. S. Siegler, Program Manager and submitted on 5 July 1978 to the United States Air Force Aero Propulsion Laboratory, Wright-Patterson AFB, Ohio, Mr. Phillip G. Colegrove, Project Engineer, in fulfillment of data requirements of Contract F33615-74-C-2013.

The report summarizes the results of studies, designs, fabrication, developmental test, acceptance test and delivery of 6000 Shp fast start turbines (< one second) driven by a monopropellant hydrazine gas generator. The work was conducted over the period of 2 January 1974 through 28 February 1978. This report also documents significant fast start gas generator advancements which were accomplished as peripheral requirements leading to successful fast start turbine development.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

SECTION I

INTRODUCTION

1. PROGRAM OBJECTIVES

Primary Demonstrable Objective

The ultimate objective of this program has been to design, fabricate, develop, demonstrate acceptance test and deliver efficient (5.0 lbs/hp-hr)[1], lightweight (250 lbs) turbines, powered by a hydrazine-gas-generator, capable of accelerating a 0.255 slug-ft^2 inertial load from rest to 29,000 rpm at 6000 shaft horsepower within 0.85 seconds.

Related Objectives

1. As a prerequisite to successful conduct of the turbine development program, it has been necessary to develop a safe, reliable, repeatable and durable fast-start hydrazine monopropellant gas generator.

2. As a prerequisite to longer range planning, it was specified that "optimum design" data be produced for the following:
   (a) Optimum Flightweight Gas Generator - To produce the preliminary flightweight design configuration for a staged hydrazine gas generator.
   (b) Optimum Flightweight Gas Turbine - To produce the preliminary flightweight design configuration for a gas turbine with extended operational life capability.

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[1] The MK15 turbine Specific Propellant Consumption (SPC) criteria of 5.0 lbs/hp-hr applies to subsonic rotor turbines E3-1 and E3-2. During 1975, test of the E1 supersonic rotor turbine exhibited a characteristic SPC of 4.5 lbs/hp-hr as noted herein and in references [4] and [20]. Numbers is brackets designate references at end of report.

2. CONCLUSIONS AND RESULTS

Fulfillment of Objectives

a) The resultant Mark 15-E3 turbine shown in Fig. 1 was demonstrated to be fully responsive to the objectives of the program.

SUMMARY OF OBJECTIVES & RESULTS

	POWER	SPEED	SPC	START
Objectives	6000hp	29,000 RPM	5.00 $\frac{\text{lbs}}{\text{Hp-Hr}}$	< 1.0 sec
Results	6000hp	29,000 RPM	5.04 $\frac{\text{lbs}}{\text{Hp-Hr}}$	0.85 sec

An average performance efficiency of 5.04 lbs/hp-hr was obtained from the sixteen E3 series tests which achieved target speed and power. Furthermore, speed capability was demonstrated to 31,813 rpm and power levels of at least 6226 SHP were recorded. Based on the .225 slug-ft^2 inertial load criteria, fast start duration was determined to be .85 seconds. More than 50 tests of the final blading configuration (Fig. 2) were conducted totaling approximately 956 seconds of powered operation.

There were 39 gas generator powered tests of the E1 and E3 turbines (23 and 16 tests, respectively) totaling 255 seconds (184 and 71 seconds, respectively), in the hot fire mode. Eleven E2 tests were conducted using compressed and heated facility air to verify blade performance. All E2 and E3 tests were conducted with a dynamometer as the shaft power loading device.

b) The multistage thermal decomposition fast start development gas generator was used to conduct more than 300 hot fire tests during this program and ultimately evolved into a very reliable, repeatable test device.

c) The "optimum" gas generator preliminary design objective was fulfilled early in the program (September 1974) before further experimental advancements of 1976 and 1977 could be applied (particularly in the areas of catalyst pack construction). A separate follow-on program was conducted to develop a high performance flightweight gas generator.

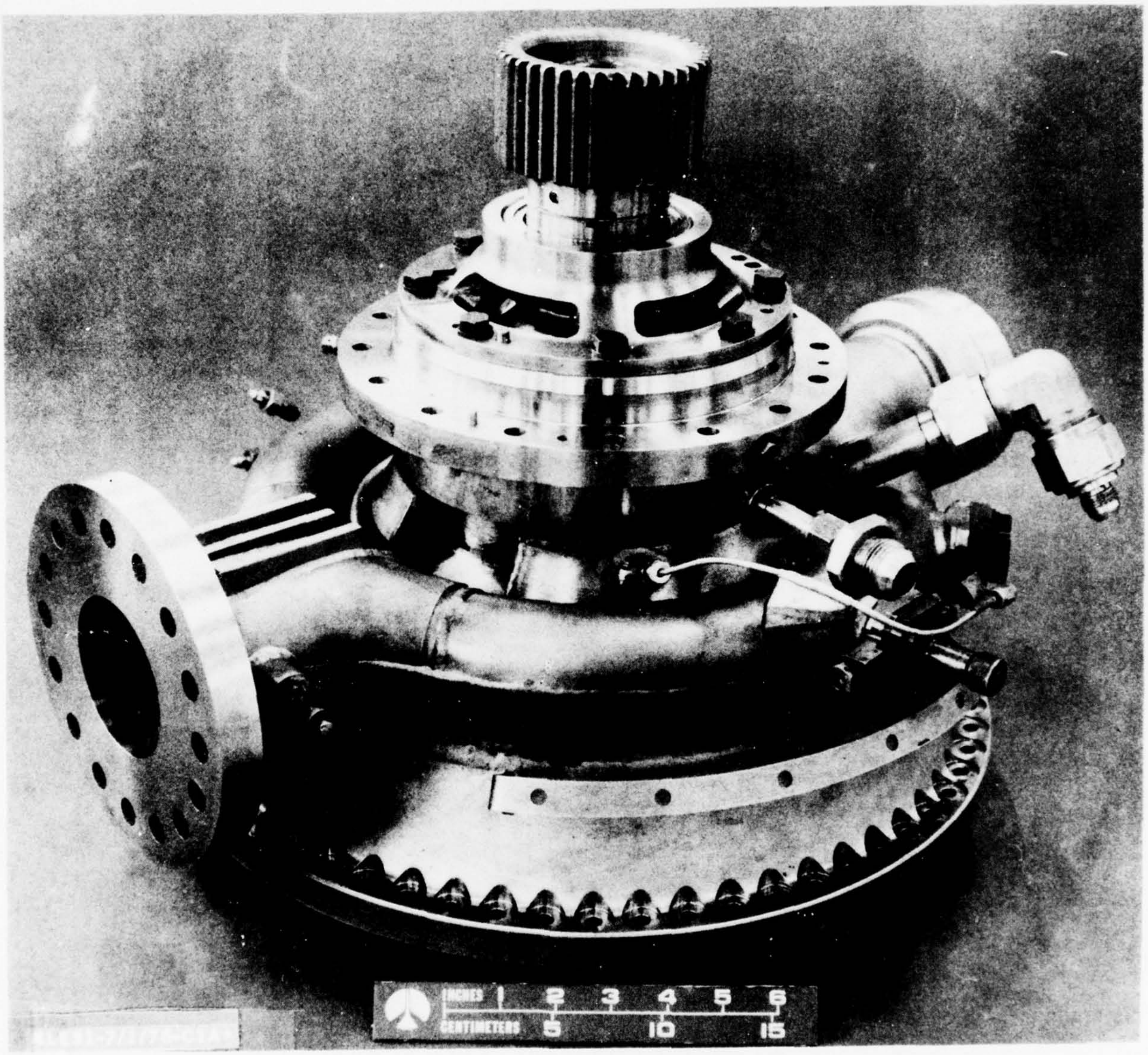


Figure 1. Fast Start Turbine Photo

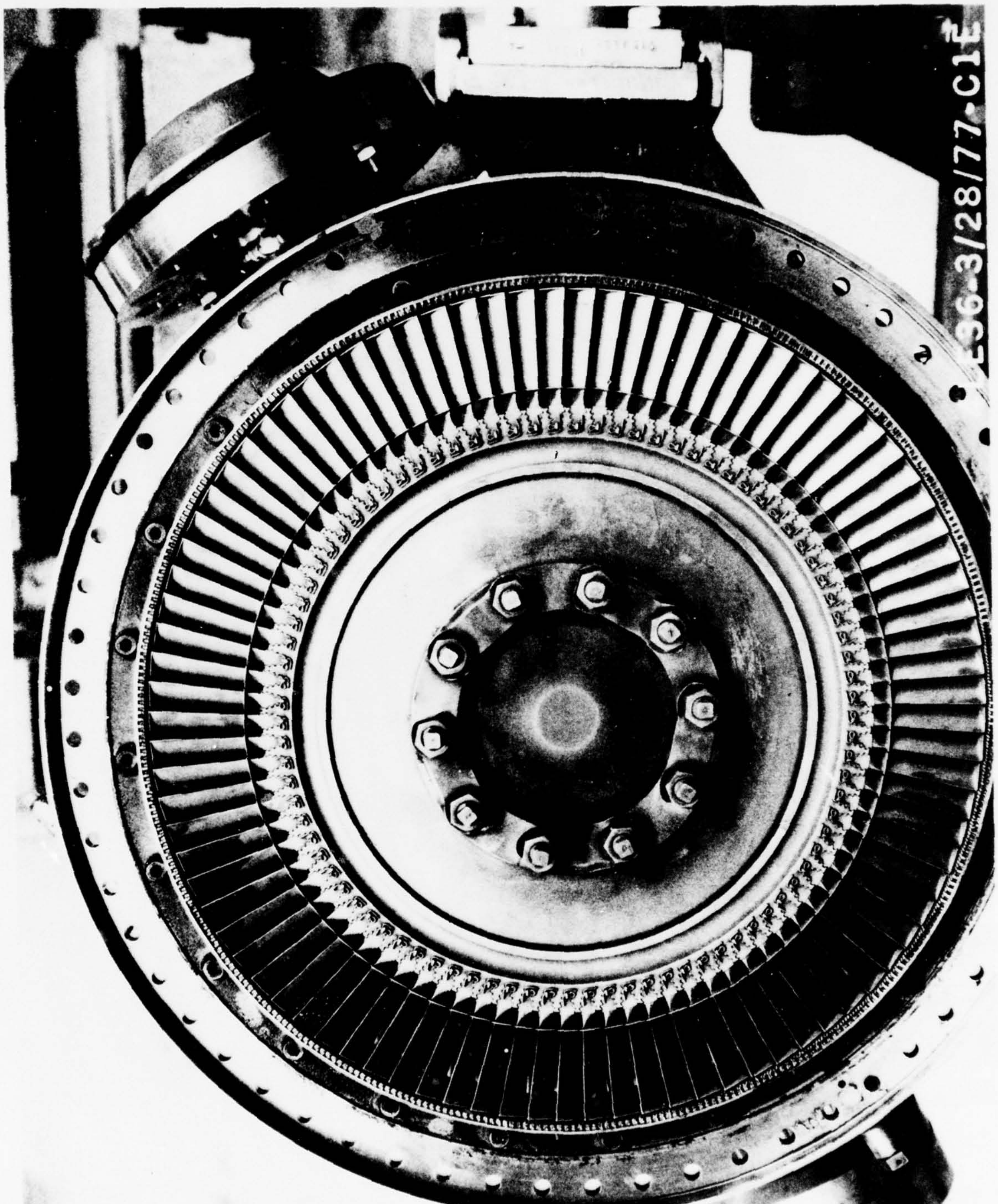


Figure 2. Turbine Rotor Blade Photo

d) The "optimum" turbine preliminary design objective was also fulfilled early in the program (May 1975). Subsequently, a number of turbine wheel and bearing design improvements were recognized and demonstrated. The key "optimum turbine" design recommendation - for a modified inlet manifold - was not within the scope of this program. The hardware used for test in this program came from prior programs where the design life criteria and the inlet design temperature were below that required here.

Recommended Action

a) The positive results of this program should be carried forth to demonstrate successful integration of a fast start power module with an appropriate gas generator at the 6000 SHP level.

b) Action should be taken to relieve the predicted turbine manifold low cycle fatigue constraint and to verify life predictions for the complete turbine. This activity would involve design and fabrication of a new and more conservative replacement manifold and low cycle fatigue test of the resulting turbine assembly.

c) Action should be taken to further modify the final E3 configuration by elimination of the separate axial thrust bearing with modifications to allow the main radial bearings to carry the minimized thrust load. This action would simplify the turbine and APU assembly and serve to increase delivery power and slightly reduce inertia of the turbine leading to improved response.

d) Studies and preliminary designs should be formulated for scale-up from the 6000 SHP level to higher power level systems with the particular objective of identifying the most cost effective and minimum risk solutions.

3. SYSTEM DESCRIPTION

All tests of this program were conducted at Rockwell's Los Angeles Division Thermodynamics Laboratory, Cell 106, as shown by Fig. 3.

A schematic drawing of the overall test installation as used to demonstrate fast start characteristics is shown as Fig. 4. Figure 5 describes the conversion to performance testing where a water brake dynamometer is substituted for the inertial flywheel assembly. A pressurized facility hydrazine tank is used to deliver propellant through facility lines, valves and sensors to the three stages of the gas generator. Gases from the generator drive the turbine and are exhausted through a 30-inch diameter facility ejector vacuum system. The nominal 29,000 rpm turbine shaft speed is reduced through a 3.2:1 gearbox to about 9000 rpm to facilitate adaptation of the two facility load devices.

Gas Generator

The fast start hydrazine monopropellant gas generator used for turbine development is a three-stage device mounted vertically with flow directed downward into the turbine inlet manifold as indicated in Fig. 3. Hydrazine flow is distributed between the pilot, first stage, and second stage in the approximate percentages of 1 percent, 8 percent and 91 percent, respectively. The pilot stage uses a 20 to 30 mesh size Shell 405 spontaneous catalyst pack to provide a continuous hot gas ignition system for the unpacked first stage reactor. The output of the first stage then serves a similar function for the downstream second stage reactor. Full cone spray pattern atomizing nozzles are used to inject hydrazine into the unpacked chambers. A thermal control catalyst bed is located within the aft section of the second stage. The pilot stage consists of an injector and stand-off manifold to spray hydrazine into the catalyst bed.

The first stage is a combustion chamber fed by the hot gas from the pilot section. Four spray nozzles introduce additional propellant into the stage. Two upstream nozzles introduce approximately 20 percent of the flow near the ignition source. These nozzles were canted toward the downstream position. The remaining 80 percent of the first stage flow is injected radially through two nozzles at a downstream location. The combustor itself is 2.68 inches in diameter with an end plate containing a rounded entrance orifice.

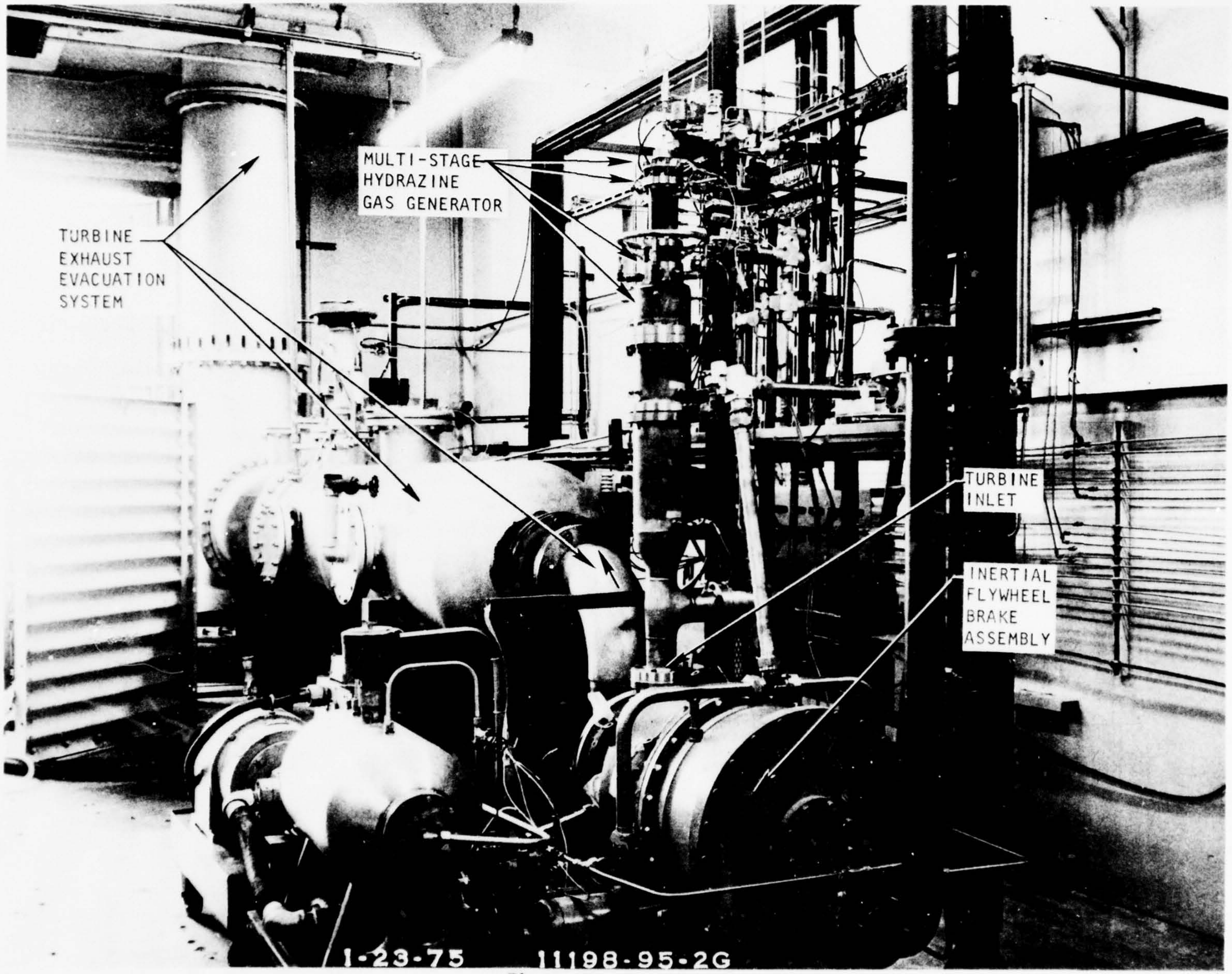


Figure 3. Facility Installation

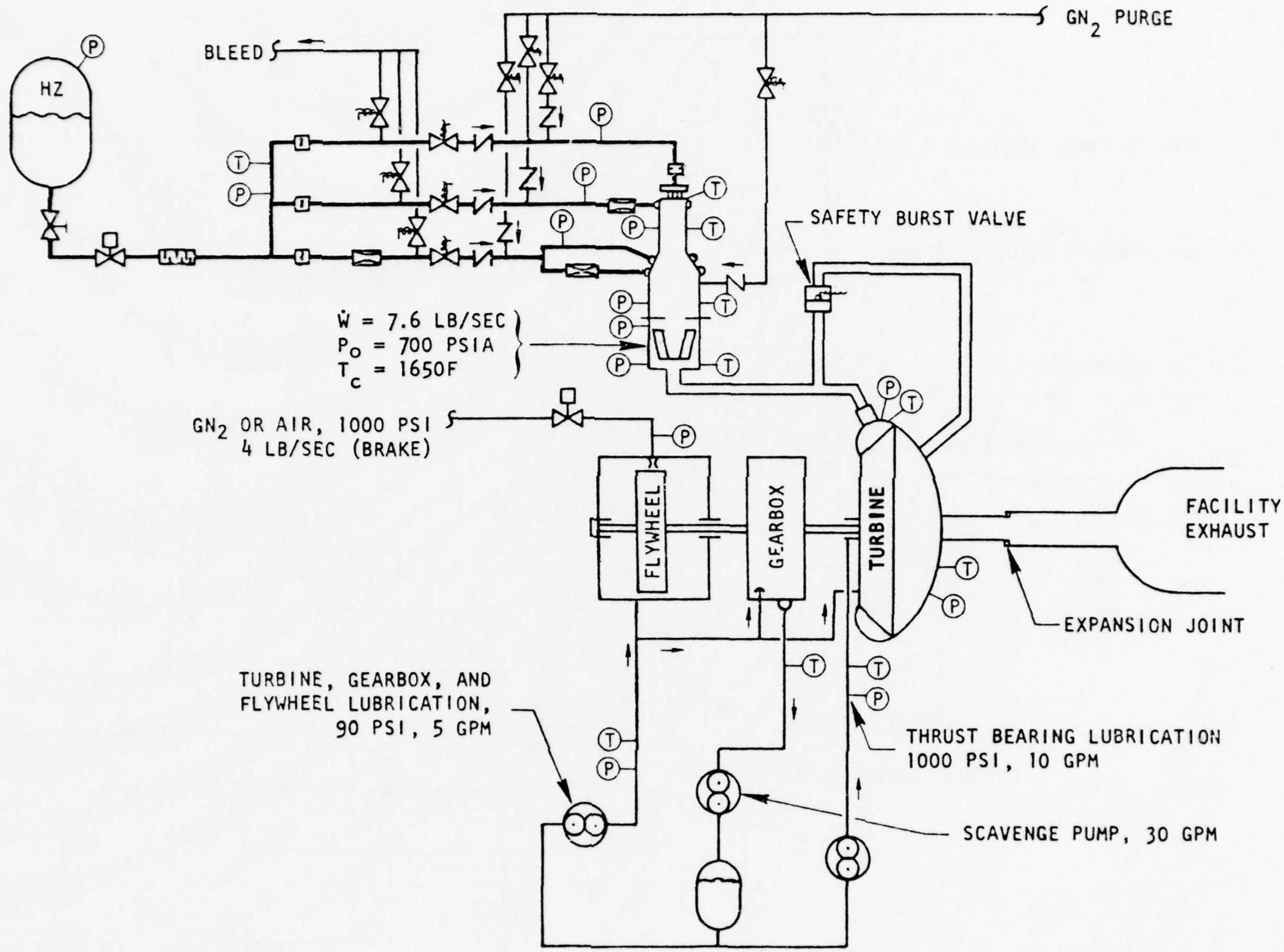


Figure 4. Acceleration Test Schematic

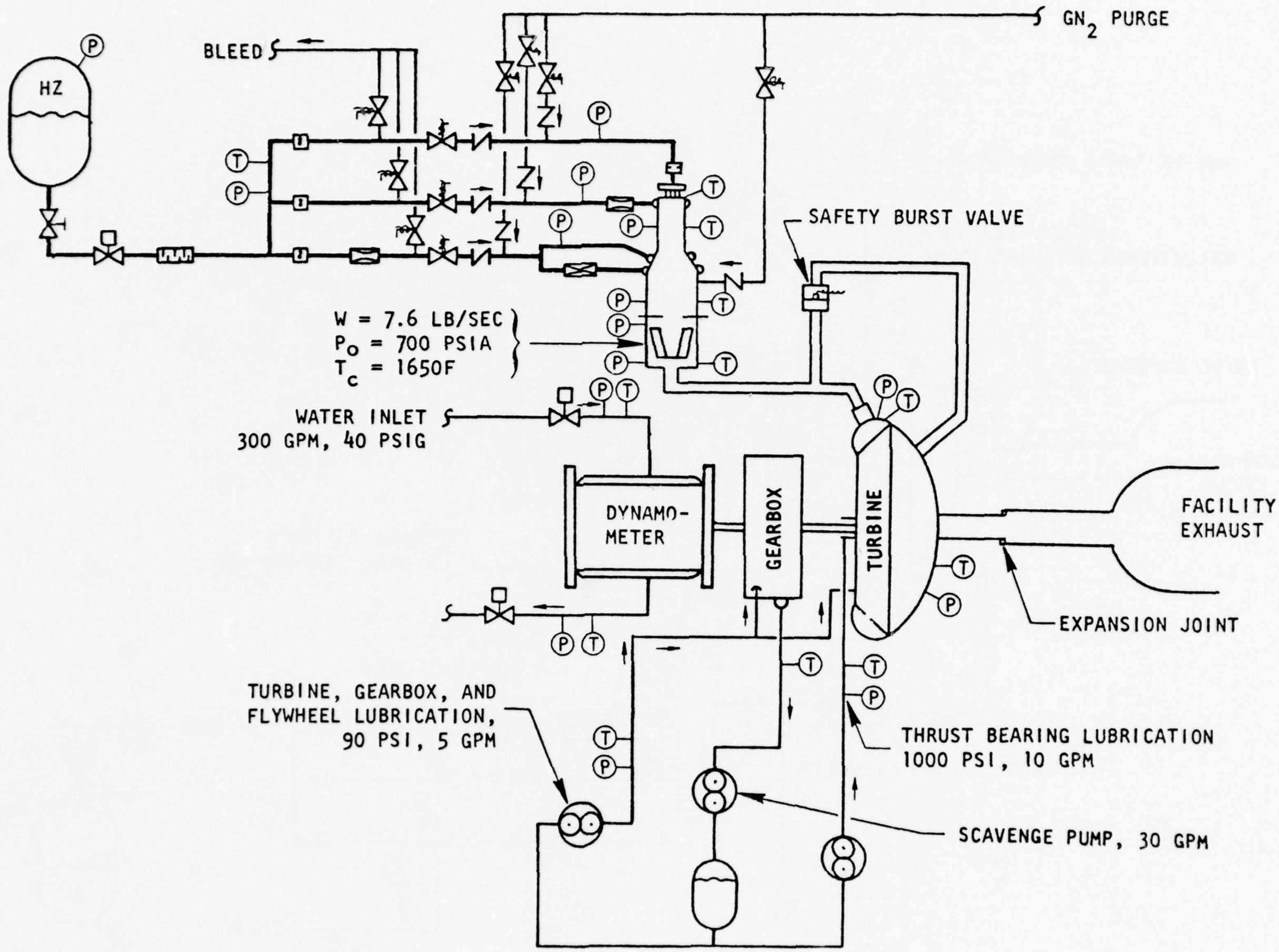


Figure 5. Performance Test Schematic

The second stage, 6 inches in diameter, has 8 injection orifices. Four orifices were located in the diverging entrance end of the combustor. These spray nozzles deliver 20 percent of the second stage flow. The remaining four orifices are downstream in the straight wall section.

The thermal control pack contains fine grain 8-12 mesh shell X-4 catalyst to obtain a high degree of decomposition of the NH_3 component of the exhaust gases. The catalyst is held in a conically shaped INCO 600 wire screen carrier which slips into the aft spool of the second stage of the gas generator.

Gas Turbine

The final Mark 15-E3 model turbine which was developed during this program is a two-row velocity compound axial flow device as shown in Fig. 6. The gas inlet manifold is a radial approach split flow configuration in the tested device because existing rocket engine turbopump hardware was used without modification. A 37-nozzle inlet ring is welded to the manifold to direct gas flow axially through the first stage rotor wheel blades. Subsequently, flow passes through a stator blade ring and finally through a second stage rotor. Both rotors are bolted together with 10 studs which pass axially between concentric curvic coupling piloting rings between the two stages and between the first stage and its mating power delivery shaft. The overhung wheel stack is supported by a set of two ball bearings between the power shaft and the housing.

Load Absorption Equipment

A 3.2 to 1.0 speed reduction gearbox was designed to support the gas turbine and adapt to either an inertia simulator flywheel or to a water brake dynamometer depending upon whether the test objective was to demonstrate fast start or power delivery. The flywheel assembly consists of a basic .195 slug-ft^2 rotor (referred to 29,000 rpm) and two attachment discs designed to allow increase of inertia to either .281 or .391 slug-ft^2. The specified design inertial load of .225 slug-ft^2 was chosen after this flywheel assembly was developed. The primary flywheel rotor is also fitted with Terry turbine slots cut into its outer periphery so that gas nozzles in the outer wall of the flywheel case can be used to slow the system quickly when desired. The

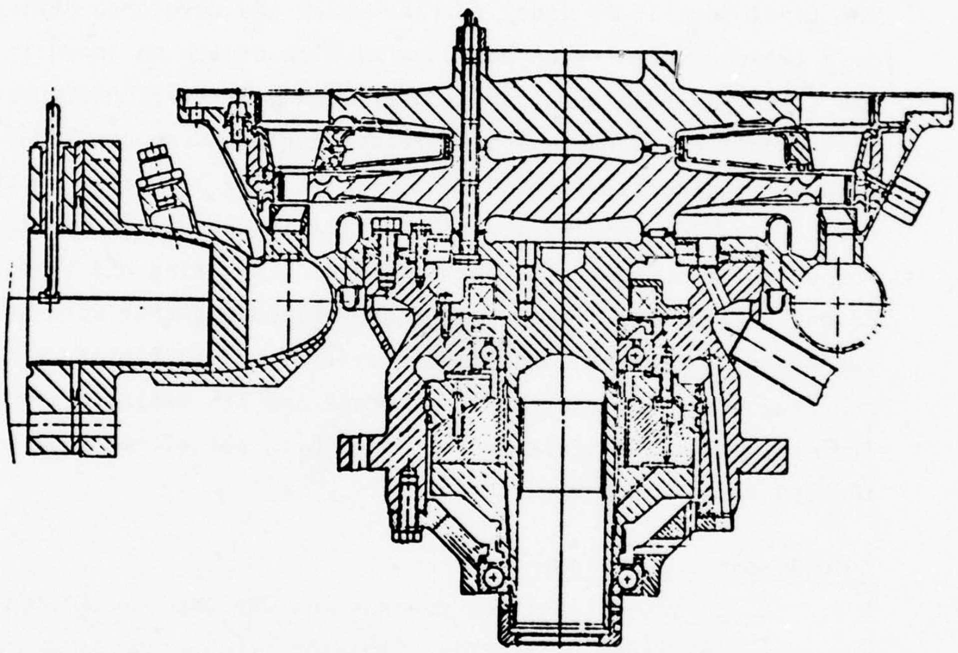


Figure 6. Test Turbine Assembly

flywheel assembly can be replaced by a water brake dynamometer. In the dynamometer water flows between rotor and stator surfaces resulting in a reaction shear torque on the housing which is proportional to power and measured by means of a strain gage load cell.

System Controls

Sequencing of the system valves is controlled by an automatic solid state sequencer which allows presetting of the start events and duration. A comparator monitors key parameters and provides for automatic shutdown should a redline value be exceeded. Prior to each test setup, the sequence conditions are specified, and the controls are set accordingly. The sequence set values are verified pretest by recording the signals on the oscillograph and monitoring an oscilloscope.

The sequence is depicted by the diagram shown in Fig. 7. The test is initiated by arming the sequencer and signaling the pilot stage valve open. When the pilot bed output temperature obtains a selected value (1200°F), the first stage valve is signaled open. The second stage valve is signaled open at a preselected time after the first stage signal, and a duration timer determines the duration of the test. Two comparator circuits are used to monitor the first and second stage chamber pressures. These monitors are set to be activated approximately 50 milliseconds after the individual stages should have normally obtained minimum pressures. If the minimum pressures are not obtained in the allotted time interval, or the pressures decay below the minimum values, the test is automatically terminated.

In the normal shutdown sequence, the second stage valve is signaled close, and the valve closes approximately 20 milliseconds after electrical signal. When the second stage chamber pressure decays below the minimum value, the first and pilot stage valves are signaled close. The lockup purge systems are manually turned on prior to the test and left on. The purges automatically stop flowing as line pressures rise above the checked off purge pressure, and the purges will automatically flow on shutdown as pressures decay. This normal shutdown sequence is used for a duration cut or minimum chamber pressure out. During tests with the inertia simulator, the gaseous nitrogen brake is manually operated posttest.

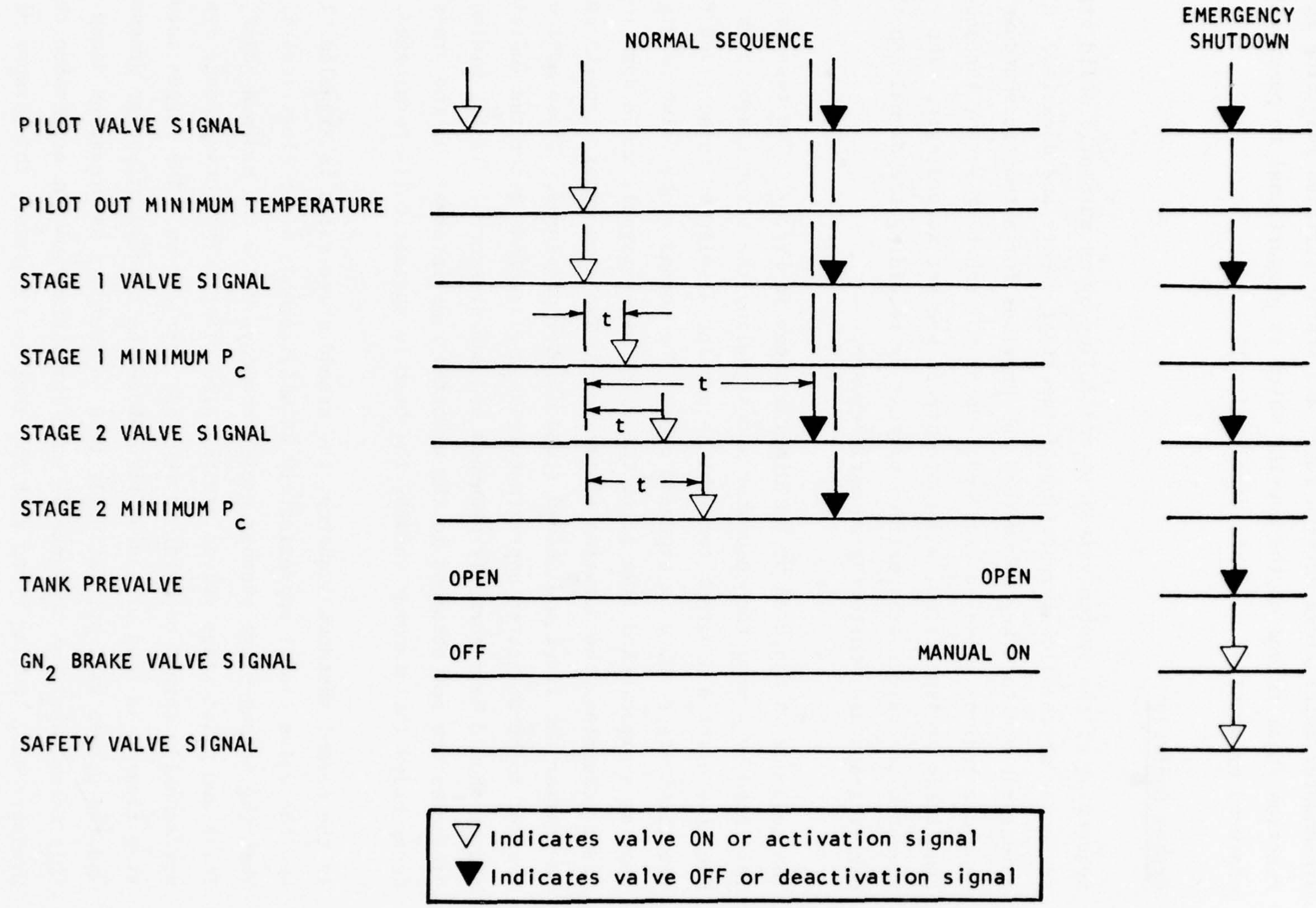


Figure 7. Controller Sequence

Two comparator circuits monitor the turbine speed and provide the capability for a normal and emergency shutdown procedure. The first speed monitor is set to a conservative value, and if an operspeed condition is detected, a normal cutoff sequence is used. The second speed monitor is set at a higher value, and if this higher overspeed condition is detected, an emergency shutdown sequence is used. This emergency sequence signals the gas generator valves closed, closes the hydrazine tank prevalve, triggers the safety burst valve, and applies the gaseous nitrogen brake.

The dual speed monitoring system is particularly applicable to the acceleration tests. The first speed monitor would be normally used to cut the test such that the turbine obtains the target speed. Should the gas generator fail to shutdown for any reason, the turbine would reach the second speed cutoff. This cutoff sequence would prevent the turbine from running away by dumping the turbine upstream pressure through the bypass system, shutting off the gas generator propellant supply.

Exhaust Vacuum System

All tests are conducted with the gas generator/turbine exhausting into a vacuum system. The vacuum in the exhaust system is provided by an Ingersoll-Rand single-stage, air-to-air ejector which is capable of continuous operation with a secondary air flow of 10 lb_m/sec at 85,000 feet altitude. The turbine exhaust gases are scrubbed in the exhaust duct with water from water spray nozzles prior to being exhausted to eliminate discharge of ammonia into the atmosphere.

4. DATA AND CHARACTERISTICS

Turbine Genealogy and Experience

The end product of this program is fast start turbine Mark 15-E3-2, which was developed from the baseline Mark 15 production rocket turbine as described by Table 1 which lists the several interim steps. The 4 MW-APU to Mark 15-EO turbine step was accomplished prior to this contract. Table 1 also summarizes the parametric test experience of these turbines and their accumulated test history.

Turbine Performance

The performance which has been demonstrated by each of the several turbines during this program has been normalized to the design conditions of the specification and summarized in Table 2.

Turbine Physical Changes

The design characteristics of the E3-2 turbine are represented by Rocketdyne drawing XEOR 943562. Table 3 describes the steps from EO to E3-2 for clarification of the key design changes which characterize each of the turbines tested during this program.

TABLE 1

TURBINE GENEALOGY AND EXPERIENCE

DESIGNATION	MARK-15F	4 MW - APU	MARK 15-EO
Design	Production	Experimental	Experimental
Application	J-2 Rocket	APU System	FST Transient
Drive Gas	48.5% O_2, 51.5% H_2	85% N_2H_4, 15% H_2O	N_2H_4
Power, SHP	8749	6000	ND
Stall Torque, ft #	ND	ND	ND
Speed, RPM	28,266	26,000	~23,000
Inlet Temp, °F	1296	1250	1603-1640
Inlet Pr, Psia	732	616	469-497
Flowrate, #/sec	7.6	12.39	9.9-9.8
Pr. Ratio	7.45	12.0	13.8-13.9
SPC, #/HP-HR (At DSN Point)	3.13	7.44	ND
Type of Tests	DNA		Fast-Start Transients
No. of Tests	-		8
Total Run Time, Sec	-		6.18
Load Device	LH_2 Pump		Flywheel

TABLE 1 (CONTINUED)

MK-15-E0	MK-15-E1	MK-15-E2	MK-15-E3-1
Experimental Facility C/O	Experimental Performance	Experimental E3 Perf. Pred.	Exp. Prototype Deliverable
N_2H_4	N_2H_4	Air	GN_2
167-4273	64-5889	90-1144	12-99 hp
ND	ND	ND	ND ft #
6,308-24,318	6232-29,785	5600-14,200	1,438-5,591 rpm
991-1559	1552-1738	9-1013	13-69 °F
3-442	554-732	51-278	27-102 psia
.9-8.0	5.87-7.93	1.60-7.42	No Data #/sec
14.3-33.7	32.6-40.1	41.7-49.6	1.8-6.5 Pr. Ratio
6.60	4.5-4.6	18.5-22.3	DNA #/HP-HR
DYNAMOMETER CHECKOUT	STEADY-STATE PERFORMANCE	STEADY-STATE PERFORMANCE	INSTRUMENTATION CALIBRATIONS
17	23	11	13 tests
87.5	183.5	735.0	143.1 sec
DYNAMOMETER	DYNAMOMETER	DYNAMOMETER	DYNAMOMETER

TABLE 1 (CONTINUED)

MK-15-E3-1	MK-15-E3-2	MK-15-E3-2	
Exp. Prototype Deliverable	Exp. Prototype Deliverable	Exp. Prototype Deliverable	
N_2H_4	GN_2	N_2H_4	
5431-6229	ND	4944-6226	SHP
1017-1263	ND	ND	ft-#
0(1)-29,840	4350-6090	26129	RPM
1272-1484	ND	1484-1547	°F
668-723	ND	625-729	Psia
8.22-8.62	ND	6.86-8.38	#/sec
40.3-42.8	ND	39.2-43.1	Pr. Ratio
4.9-5.5	ND	4.7-5.1	#/HP-HR
Acceptance Integrity/Perf	Instrumentation Calibrations	Performance Mapping	
7(1)	10	9	Tests
33.8	7.0	37.4	Secs
Dynamometer	Dynamometer	Dynamometer	

(1) Includes two stall torque tests 0 rpm

TABLE 2 TURBINE PERFORMANCE

	E0 TURBINE	E1 TURBINE	E2 TURBINE		E3-1 TURBINE	E3-2 TURBINE	DESIGN SPEC	UNITS
SPEED	25,597	29,000	29,000	29,000	29,000	26,129 29,000 31,813	26,100(min) 29,000(nom) 31,900(max)	RPM
POWER	5,000	6,000	960	6,000	6,000	6,000	6,000	SHP
FLOW	9.6	7.58	4.95	8.16	8.50	8.40	8.33 (nom) 9.17 (max)	LBS/SEC
INLET PRES	566	700	220	710	710	710	710	PSIA
INLET TEMP	1650	1650	800	1450	1450	1450	1450	°F
SPC	6.91	4.55	18.56	4.90	5.10	5.04	5.0 (nom) 5.5 (max)	LBS/HP-HR
PROPELLANT	N_2H_4	N_2H_4	AIR	[CALC N_2H_4]	N_2H_4	N_2H_4	N_2H_4	-

TABLE 3

TURBINE PHYSICAL CHANGES

MK 15-	PART NUMBER	STATUS
E0	XEOR939921D1	IN STORAGE
	Replace wheels, stator and nozzle to optimize performance for N_2H_4 (rather than H_2O_2)	
	Pilot on curvic couplings rather than studs	
E1	XEOR941470	PARTS USED
	Redesigned blading for subsonic rotor (was supersonic)	
E2	XEOR941470D20	PARTS USED
	Revised from integrally bladed wheels to fir-tree-root blades set into wheels	
	Reduced design inlet temp. by 200°F (to 1450°F)	
E3-1	XEOR944120	DELIVERED
	Reduced from 41 to 37 inlet nozzles	
	Increased rigidity of rotor stack	
E3-2	XEOR943562	DELIVER TO STORAGE

5. PROGRAM SUMMARY

A 50-month Fast Start Turbine (FST) system program was conducted by the Rocketdyne Division of Rockwell International during the period of 2 January 1974 through 28 February 1978, as shown by Figure 8. Activities were initiated in 1974 to conduct tradeoff studies, perform preliminary design and to conduct component development tests with the aim of demonstrating the technical feasibility of a highly efficient hydrazine driven 6000 Shp gas turbine. As tradeoff studies proceeded, a decision was reached to expand the program scope to include development of a companion lightweight, multistage, fast start hydrazine gas generator for earlier demonstration of turbine response under inertial loading. A design task was also added to redefine the developmental gas generator as an "optimum" flightweight design concept.

By the end of the 14th program month, the developed generator had been utilized successfully with an interim (less efficient) turbine (MK15-E0) and had demonstrated the fast start objective. With this added confidence and experimental background, the Phase II preliminary (MK15-E1) turbine design continued, resulting in hardware for test in the 21st program month.

Meanwhile, fabrication of a second turbine was authorized on 5 May 1975 to meet Air Force Weapons Laboratory (AFWL) specifications for use in a prototype APU program. For this deliverable turbine AFWL selected a turbine inlet temperature of 1450°F. Analytical studies were included in the new task to determine whether the MK15-E1 configuration (supersonic rotor) turbine would operate properly some 200°F below its design point and fabrication of the inlet nozzle assembly and wheels for the second turbine was temporarily constrained.

In late July of 1975, a decision was reached to follow a more conservative path with the second turbine design. The revised turbine was designated MK15-E2 and its design was completed by Rocketdyne during August and September of 1975.

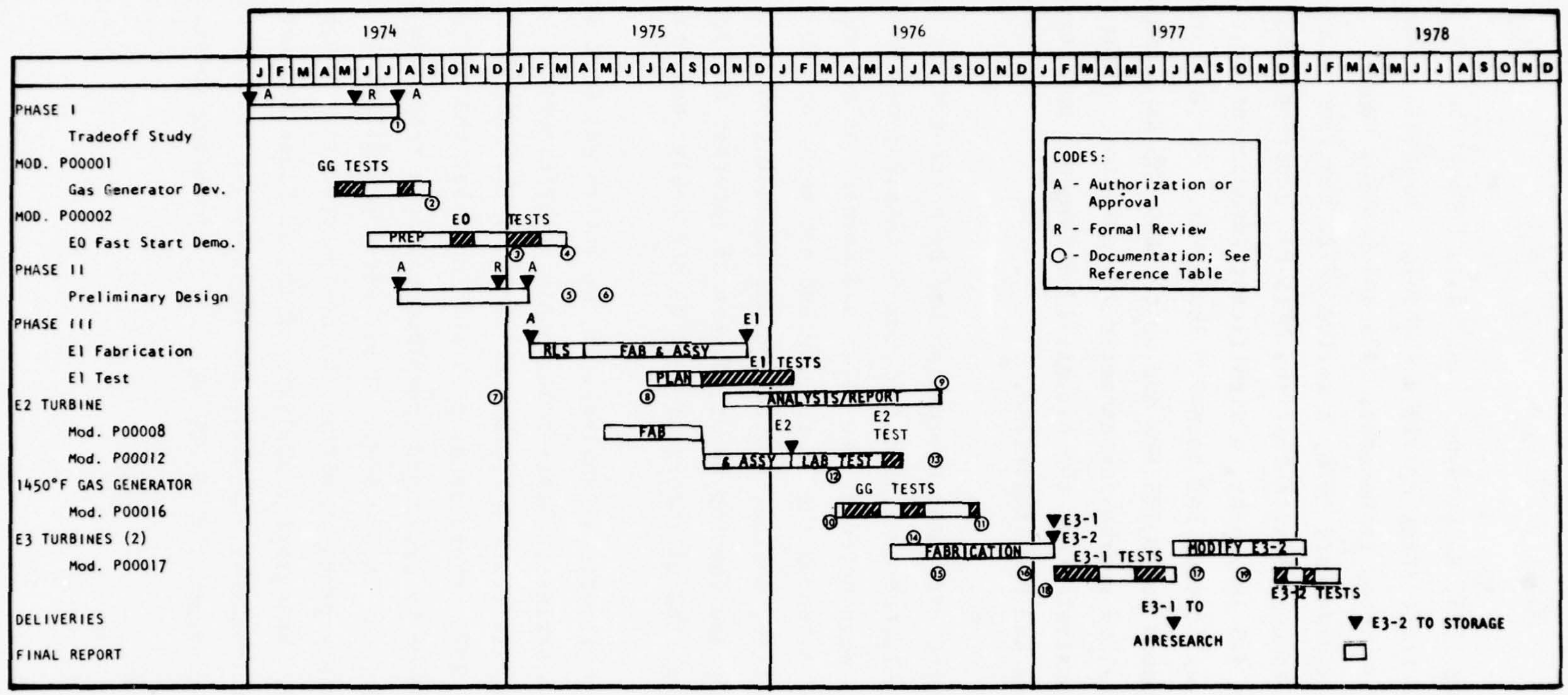


Figure 8. Program Schedule

On 25 September 1975, Rocketdyne received authorization for fabrication of the MK15-E2 first stage nozzle and wheels. Meanwhile, the MK15-E1 assembly was completed in December 1975 and testing began. During December 1975 and January 1976, a series of 23 hot fire turbine tests at 1650^{o}F served to demonstrate that the MK15-E1 supersonic turbine efficiency was 4.5 lbs/hp-hr, a significant advancement over the MK15-E0 turbine whose efficiency had been 7.0 lbs/hp-hr. The E1 turbine was operated at speeds to 29,785 RPM and up to 5889 Shp as measured through the newly installed gearbox/dynamometer installation. These results served to demonstrate that the essential performance objectives of the initial program had been fulfilled.

In February 1976, after the E1 turbine had been removed for posttest inspection (to remove contamination from GG catalyst bed failure), fatigue cracks were noted at the roots and corners of several first stage blades. Although the E1 turbine had not been intended as an optimum, long life, device, a detailed understanding of the vibration mode, frequency and forcing function were of interest and action was taken to analyze the E1 rotors as well as the newly completed E2 rotors.

In particular, vibrational analysis of the rotors was most significantly benefited by a series of laser holographic interferometric vibration frequency analysis tests performed at AFAPL by Dr. James C MacBain. During March, 1976, these studies established that the E1 turbine damage was caused by prolonged operation within a resonance regime at reduced speed (near 20,000 RPM). There were no indications of E1 turbine vibratory problems within the design operating range of 26,100 to 31,900 RPM. Holographic analysis of the E2 blade indicated that the blade natural frequency was above the analytically calculated value and unacceptably close to the 29,000 RPM design operating point.

The design solution which was selected was to retain the conservative E2 aerodynamic blade design but to use a shrouded fir tree root (non-integrally bladed) rotor configuration. Further AFAPL holographic test of the E2 blade modified to an E3 configuration (with the shroud and fir tree attachment). These tests predicted a 12.5% margin between the forcing function at 29,000 RPM design speed and the 22,282 H_2 natural frequency of the blade (which is encountered at 32,604 RPM).

Concurrently, planning began to test the existing E2 turbine using heated air at reduced speeds to determine its performance. As predicted, the E2 turbine performance was approximately 11% lower, consuming the equivalent of 5.0 lbs-N_2H_4/HP-HR.

On 26 April 1976 additional development of the fast start hydrazine gas generator was authorized to attain the 1450°F target gas delivery temperature for extended life benefits to the forthcoming deliverable hardware. On 15 July 1976, authorization was received to implement the plan for fabrication of E3 turbine rotor components, assembly of two E3 turbines using E1 and E2 parts (as applicable), acceptance test and delivery of the MK15-E3 turbines for APU service.

By January of 1977, the E1 and E2 turbines (with integrally bladed rotors) were replaced by the first deliverable (E3-1) turbine using the fir-tree-root method of attaching rotor blades to the wheel disc. The E3-1 turbine was delivered to the customer in May 1977, following acceptance test to 103% of rated speed and satisfaction of all integrity and performance objectives. The second E3-2 turbine was also ready in January 1977, but was thereafter assembled with 37 rather than 41 inlet nozzles to raise the upper speed capability further. By using 10% fewer nozzles the 22,282 H_3 resonance speed was forced upward to 36,178 RPM. Testing was thereafter successfully completed to over 110% of rated speed.

At the conclusion of the program, both E3 turbines and all test devices were in excellent condition. Major program milestones are shown in Table 4.

TABLE 4

MAJOR PROGRAM MILESTONES

DATE	EVENT	ACHIEVEMENT
12 Aug. 1974	Phase I study approved Phase II design authorized	Turbine and APU design characterization completed
16 Aug. 1974	Fast start gas generator development completed	9000 HP fast start N_2H_4 gas power supply capability demonstrated
21 Jan. 1975	Phase II design approved Phase III test authorized	El turbine design and test plan completed
10 Feb. 1975	Fast start intertially loaded EO turbine tests completed	.85 second fast start capability demonstrated
1 Oct. 1975	Procurement of component parts initiated for integrally bladed E2 turbine	Subsonic E2 turbine design (for more conservative power margin) selected to replace El supersonic design.
15 Oct. 1975	Assembly of El turbine completed	Optimized performance turbine ready to test
30 Jan. 1976	Test of El turbine completed	Specified 4.5 lbs/w-H, 6000 SHP, 29,000 rpm criteria demonstrated
11 June 1976	Procurement of E3 turbine parts initiated for two deliverable turbines based on E2 aerodynamics and fir tree root design.	Long-lead cast-blade turbine design and stability tests successfully concluded
19 June 1976	E2 turbine aerodynamic (cold) tests completed	Performance of extended life blading profile successfully demonstrated

TABLE 4 (CONTINUED)

DATE	EVENT	ACHIEVEMENT
15 Sept. 1976	Thermal control pack testing completed	Capability of producing 1450°F N_2H_4 inlet gas demonstrated for increased turbine life.
28 Jan. 1977	Assembly of E3 turbines completed	Deliverable turbines ready for acceptance test
13 May 1977	E3-1 turbine test and delivery complete	First deliverable turbine accepted by customer
30 Jan. 1978	E3-2 turbine tests completed	Second deliverable turbine successfully demonstrated performance and limits capabilities.

SECTION II
DISCUSSION

1. PHASE I

Tradeoff Study [1]

Objectives - The tradeoff study phase of the present program had as its major objectives, (a) to select the best lightweight system approach for providing multimegawatt power, (b) to define an optimized turbine specifically for duty with that auxiliary power system. Potential missions for this power system require that full power be delivered for relatively short lengths of time at intermittent periods within 1 second of demand. System efficiency during the overall mission is a critical factor influencing practicality.

Complete system definition required trade studies among various configurations with particular reference to meeting a 1-second start requirement by either a fast start or by an efficient idle mode of operation. The fast start system accelerates from zero to full speed and power output within 1 second of demand; the efficient idle system remains at full speed with zero power output until demand.

The aircraft-dependent efficient idle system may use a small amount of power from the aircraft to idle all or a portion of the system at full speed so that the full power demand can be satisfied within 1 second, because only a portion of the system is required to accelerate in that time. An aircraft-independent efficient idle system is also defined as one which has its own separate power and energy sources.

Criteria - The criteria for selection of the system approach are: (1) minimum wet weight; (2) capability of full power output within 1 second; (3) low development risk and (4) high reliability. A related objective of the tradeoff study is the selection of the best turbine approach to be used with the selected system. The selection of a system is dependent on the duty cycle requirements and design goals. Four separate operational duty cycles have been identified and the auxiliary power system (APS) must be designed to operate

under the most severe constraints imposed by these duty cycles. Life of the system must be 100 complete duty cycles with a complete cool-down between them. The four duty cycles were as defined in Table 5 and Figure 9

TABLE 5. CYCLE LIFE REQUIREMENTS (100 DUTY CYCLES)

DUTY CYCLE	STARTS/ SHUTDOWNS	FULL POWER DURATION, HOURS
10 seconds on 10 seconds off (3 times)	300	0.833
2 seconds on 3 seconds off (15 times)	1500	0.833
15 seconds on 600 seconds off (2 times)	200	0.833
150 seconds	100	4.167

For the fast start system, an alternator weight of 960 pounds and an inertia of 2 slug-ft^2 at 8000 rpm were used. These corresponded to goals under other Air Force-sponsored programs. The initial study consisted of synthesizing and evaluating system configurations to determine the most promising concepts. Evaluation criteria were established and a quantified rating scheme was used to compare candidate systems. The ground rules used in evaluating system and component tradeoffs are given in Table 6. They are predicated on Statement of Work requirements and related experience gained from recently completed studies and test programs.

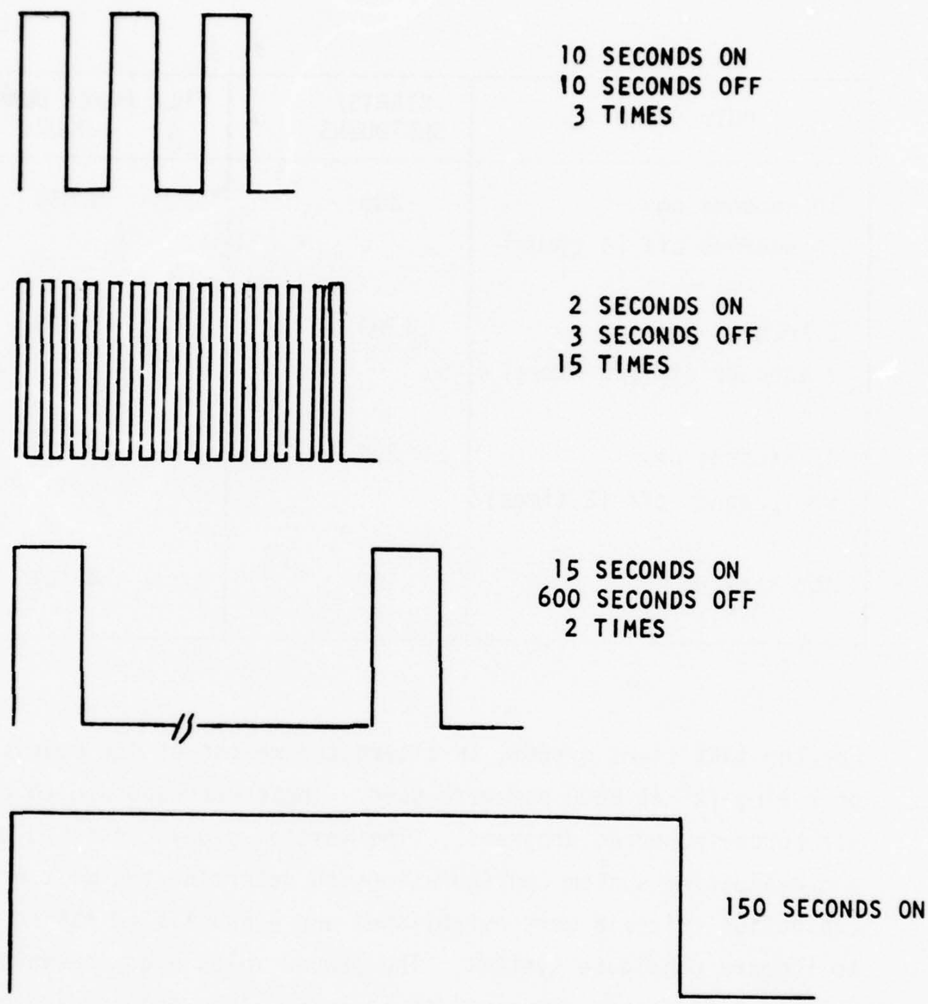


Figure 9 APS Duty Cycles

TABLE 6 TRADE STUDY GROUND RULES

Ground Rule	Rationale
Neat N_2H_4 + H_2O as required	Adjust for required gas temperature
105-percent peak turbine flow-control margin	Airborne APU contract; dynamic studies
10-percent turbine overspeed capability	Typical for APU
Gas generator throttling range of 105 to 75 percent	Deep throttling not required for any approach
Gas generator performance	Rocketdyne test experience
100-percent η_{c*}	
62-percent NH_3 dissociation	
Tankage	
6 Al-4V titanium	Parametric data available
Spherical	FSS contract
0.020-inch minimum wall	APU Design Handbook
Safety factor = 2.25 on ultimate stress	
Safety factor = 1.50 yield stress	
Relief 20 percent above operating pressure	
Tankage sizing	APU contract ground rule
5-percent residual	
5-percent ullage-load at 70 F	
15 starts for fast-start system	Most conservative assumption
Assume speed = 0 at end of 150 milliseconds for fast start and allow 0.85 second for acceleration	Provides margin under 1 second and simplifies dynamic analysis
Design guidelines	Airborne APU ground rule
AFSC design handbook (1-6), "system safety"	
Handbook DH 1-X, "checklist of general design criteria"	

Methods - A set of 13 evaluation criteria were used to select the best approach for providing the required mission power and a quantitative merit rating system was developed around these 13 criteria. The minimum and maximum merit rating values were 0 and 1, respectively, with a value of 1 corresponding to the desired objective. Merit rating values for wet weight, size and SPC characteristics were determined by dividing the lowest value for each of these characteristics by the corresponding value for each candidate system. (Thus, for example, the lightest weight system would have a weight merit rating of 1.). The response merit rating is equal to 1 minus the time between the command to open the gas generator valve and the time of alternator loading. The reliability merit ratings were calculated by dividing the unreliability of the most reliable system by the values of each candidate system.

Eighteen system combinations of options were defined, major technology areas were identified and then the list of candidates was narrowed to nine during initial screening. Table 7 indicates the results of this interim step before final evaluation began. Considered for further study were the fast-start system and idle systems utilizing either an evacuated main turbine or a clutch, with idle power provided by either hydraulic, electric, or pneumatic aircraft-dependent sources or an aircraft-independent hydrazine gas generator and turbine. Fig. 10 and 11 schematically display characteristics of two representative systems. Analytical results indicated that the best idle system is either electric or hydraulic because pneumatic system demand from compressor air was not judged acceptable. Both the clutched and the evacuated idle system approaches required development of advanced components such as a 30,000 rpm, 1000 ft # clutch or a lightweight, leak free vacuum valve for the turbine exhaust. A two-stage turbine was selected because of reduced complexity, risk and program cost relative to a four-stage turbine.

TABLE 7 IDLE SYSTEM MERIT RATINGS BY CATEGORY

IDLE SYSTEM	WEIGHT	SIZE	SPC	RESPONSE	DEVELOPMENT RISK	RELIABILITY	ALTERNATOR TECHNOLOGY	STANDBY CAPABILITY	AIRCRAFT INDEPENDENCE	MAINTAINABILITY	LOGISTICS	OPERATIONAL FLEXIBILITY	SAFETY
EVACUATED/H_2H_4	0.894	0.842	1.0	.875	0.25	0.916	0.25	0.5	0.75	1.0	0.25	1.0	0.67
CLUTCH/N_2H_4	0.958	0.907	1.0	.542	0.25	0.773	0.25	0.5	0.75	1.0	0.25	1.0	0.67
EVACUATED/PNEUMATIC	1.000	0.996	1.0	.875	0.25	1.000	0.25	1.0	0.25	1.0	0.25	1.0	0.67
CLUTCH/PNEUMATIC	0.997	0.966	1.0	.542	0.50	0.831	0.25	1.0	0.25	1.0	0.25	1.0	0.67
EVACUATED/HYDRAULIC	0.981	1.000	1.0	.875	0.25	0.727	0.25	1.0	0.25	1.0	0.25	1.0	0.67
CLUTCH/HYDRAULIC	0.983	0.970	1.0	.542	0.50	0.691	0.25	1.0	0.25	1.0	0.25	1.0	0.67
EVACUATED/ELECTRIC	0.940	0.987	1.0	.875	0.25	0.882	0.25	1.0	0.25	1.0	0.25	1.0	0.67
CLUTCH/ELECTRIC	0.974	0.966	1.0	.542	0.50	0.750	0.25	1.0	0.25	1.0	0.25	1.0	0.67

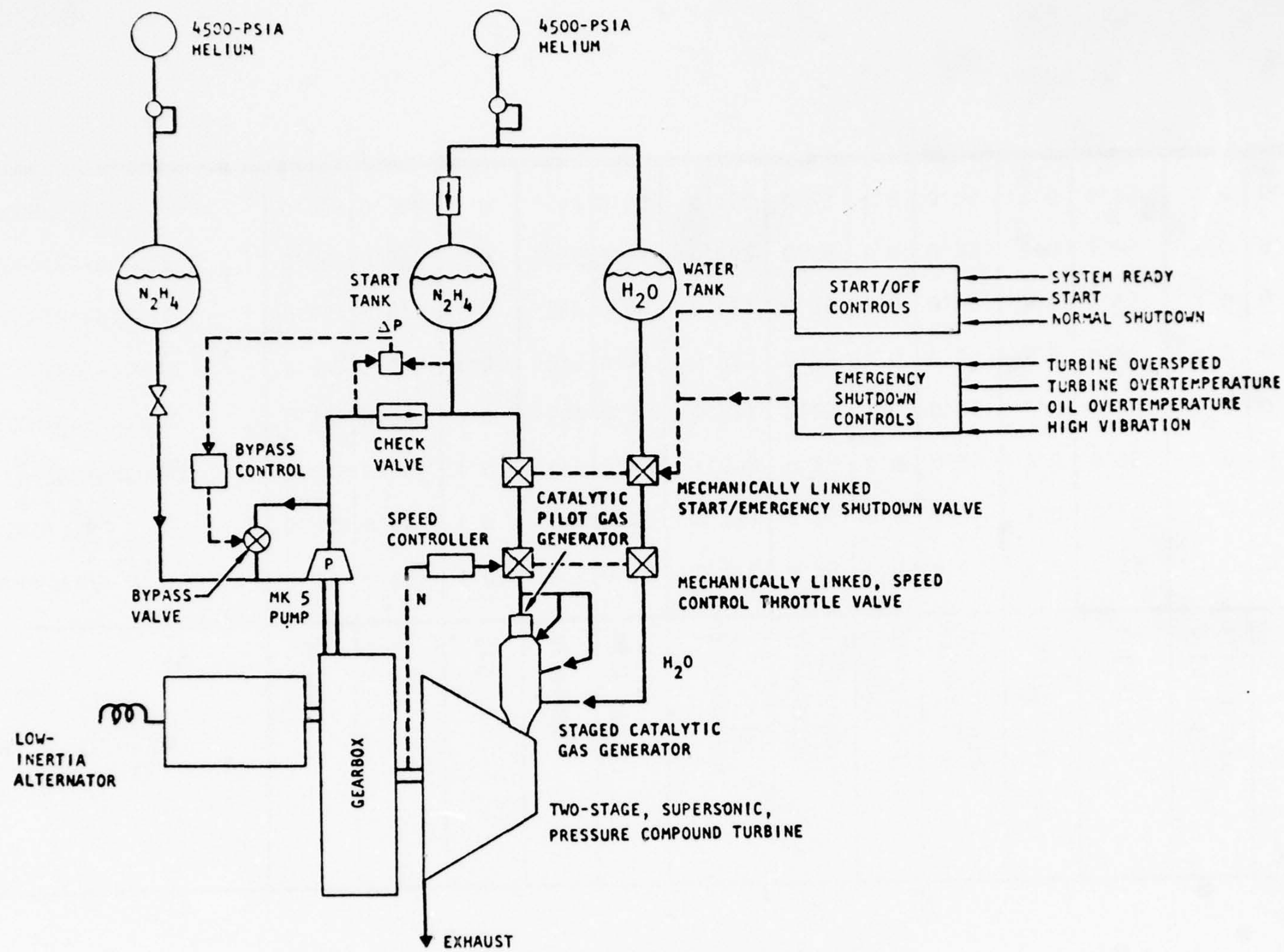


Figure 10 Fast-Start APS Schematic, Aircraft Independent

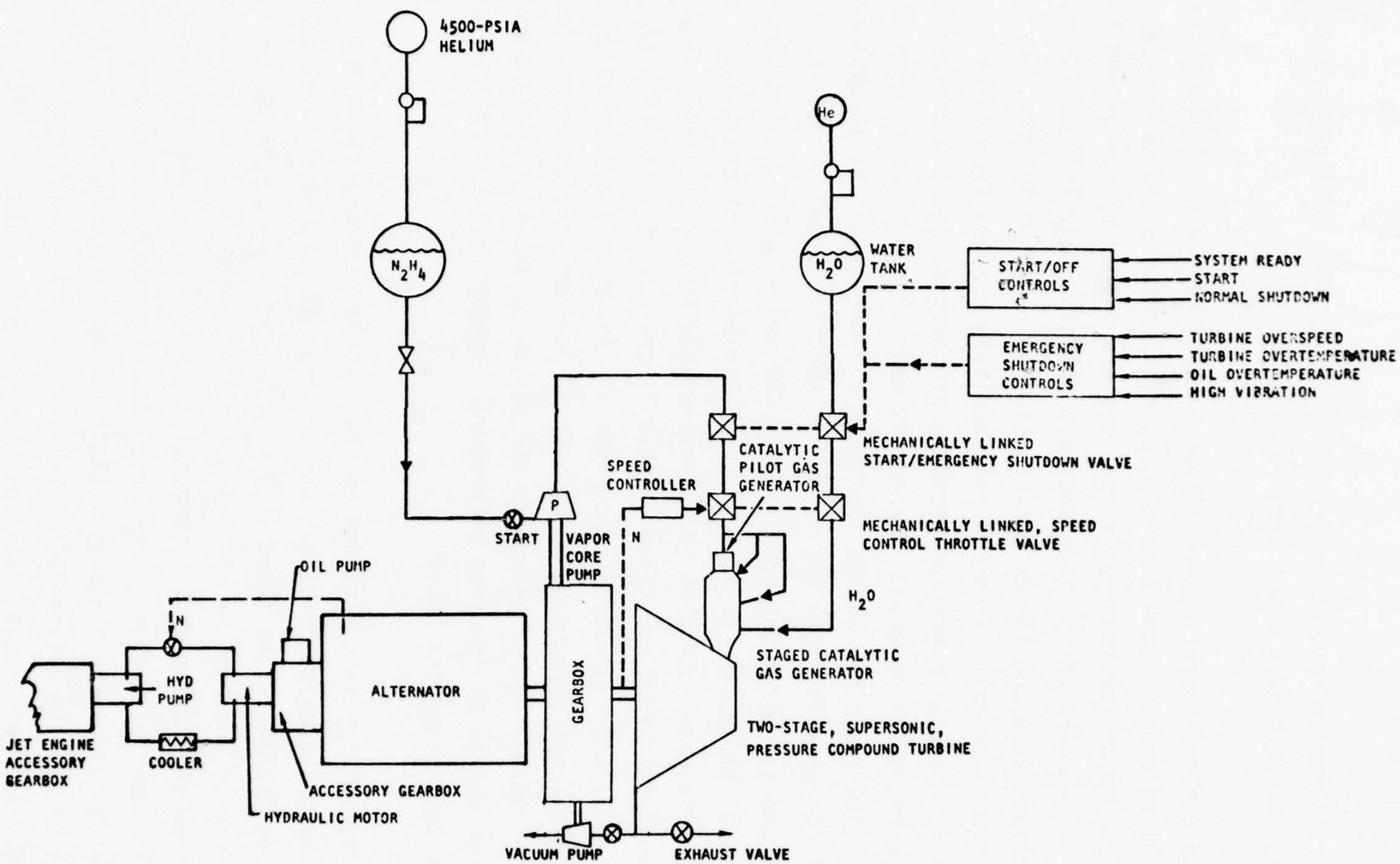


Figure 11. Efficient Idle/Evacuated/Hydraulic Drive APS, Aircraft Dependent

The analysis procedure used in selecting the optimum turbine type and turbine operating conditions is summarized in Fig. 12.

The major turbine operating conditions which were varied in the study are turbine inlet pressure and temperature, exit pressure, rotating speed and tip speed. The major turbine type variations are number of stages and type of staging as represented by stage pressure splits. Only full admission, axial flow turbines were considered because, in the range of interest, neither partial admission axial flow nor full admission radial flow turbines are acceptable. Several checks on geometry are made during the execution of the turbine performance program. A check is made to see whether, with the existing flow and blade angles, sufficient blade height or annulus area is available to pass the required flow. If not, the blade height is increased in value and an appropriate penalty for the use of a nonoptimum blade height is applied. Similarly, a check of the power bending stress induced at the blade root is made. If this is excessive, the number of blades is reduced. The blade axial width for fixed solidity increases linearly with reduced blade number, thus increasing the blade stiffness by the cube of the blade axial widths, while increasing the bending load only by the first power of the axial widths. Other geometric variations of interest include use of nonoptimum degree of reaction to produce a higher or lower Mach number relative to the turbine rotor.

The blade geometry information from the turbine performance program is next fed to the disk design program. This program is much more complex than is implied by the small box shown in Fig. 12. This program takes the turbine blade geometry and turbine operating condition data from the performance program and calculates the corresponding turbine blade heat transfer. Using this heat transfer data and the appropriate boundary temperature conditions, the program calculates the steady-state temperature gradient in the disk. For that steady-state temperature gradient, it then calculates the minimum disk thickness required to just hold the desired factor of safety on stress. Since both the steady-state temperature distribution and the stress are related to the disk thickness, this is an iterative procedure.

Because this disk design program utilizes the full strength of the materials to resist the combined centrifugal and thermal stresses in the disk, some designs (especially for lower tip speeds and lower temperatures) are found to have unrealistically thin hubs. Since such a thin hub does not provide adequate room for attaching shafts, and a thin disk may induce umbrella mode vibrations in the disk, an alternate approach to calculation of turbine weight and inertia is used. In either case, the turbine rotor weights and turbine rotor inertia are made available to the next computer program.

A third computer program utilizes the turbine inertia and the turbine torque characteristics from the turbine performance program to calculate the acceleration time for a given load inertia. The initial calculation for each sequence assumed the 2 slug-ft^2 (at 8000 rpm) alternator inertia. Since all of these systems accelerated in less than the desired 0.85 second, additional calculations were made to find the maximum load inertia which could be accelerated in 0.85 second.

Finally, the turbine weight results were combined with the operating parameters to calculate the weight of the rest of the system in the fourth program. This program provides system dry and wet weight estimates both for the fast start and for the efficient idle systems. As explained before, the optimum is generally taken to be the minimum wet weight, fast start system.

<u>Results</u> - The results of the systems tradeoff study lent emphasis to changes which were developing in objectives for the fast start turbine program.

1. Although the highest merit ranking efficient idle systems and fast start systems did not show significantly different merit rankings, it was significant that:

   (a) Both approaches required development of a lightweight alternator for airborne service which is inherently reduced in inertia so that this alternator tends to satisfy fast start system criteria

   (b) The turbines for either type of system tended to optimize at the same operating conditions, thus the same turbine could be used for either system.

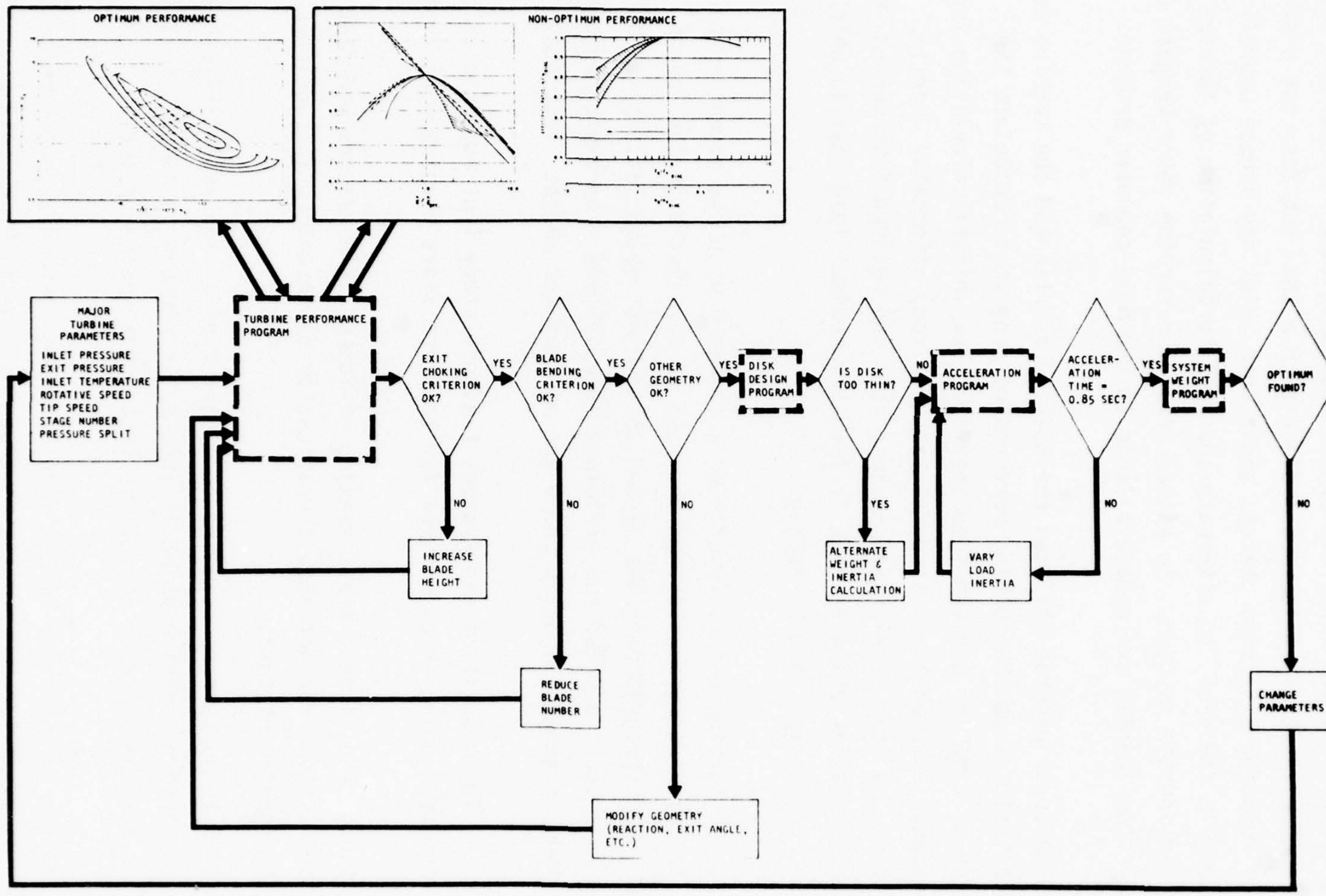


Figure 12 Turbine Analysis Procedure

2. On the basis of 1(b) above, early demonstration of the turbines fast start capability would allow subsequent attention to be focused upon turbine performance, integrity and durability satisfying either type of system.

Gas Generator Development (MOD P00001) [2]

Initial plans to use a previously developed, heavy-duty, single-stage, hydrazine gas generator for component development were modified as a result of tradeoff studies and discussions during the first three months of the program. As more specific objectives were defined, a new task was added to the program during 1974 for modification of the existing large scale single-stage IR&D gas generator taking advantage of experience with smaller multi-stage IR&D generators to obtain a relatively lightweight, fast start, moderate exhaust temperature generator. This task was initiated in compliance with contract MOD.Pooool effective 22 April 1974, to demonstrate the feasibility of utilizing the staged reaction concept for a fast start, monopropellant hydrazine gas generator. The primary objective was to determine the stage ratio where hot gas from the first stage initiates and sustains decomposition of liquid hydrazine in the second stage. Testing was continued to explore and map operating limits of the gas generator. Additional areas of investigation were injection geometry, start transients, pilot catalyst bed design, and exhaust temperature control.

The gas generator used in this program consisted of a catalyst pilot stage, first stage and second stage, and this assembly operated at a nominal chamber pressure of 590 psia and total flowrate of 8.5 lb_m/sec. The gas generator design was based on previous IR&D work, and the pilot and first stage utilized hardware from a previous IR&D effort. The design consisted of a low flowrate (1 percent of total flow) pilot stage which catalytically initiated hydrazine reaction. The pilot stage hot gas then initiated reaction in the first stage combustor (8 percent of total flow) which, in turn, initiated reaction in the second stage combustor. Thirty-two different gas generator assembly configurations were fired and variables were pilot catalyst bed design loadings and geometries, feed system modifications, second stage catalyst beds and injection geometries.

The test program was conducted in two phases during the period 8 May to 16 August 1974, and consisted of 156 tests of the gas generator including 18 tests with pilot only, 111 tests with pilot and first stage only, and 27 tests with all stages operating. Total successful hot fire time for all stage operation was 474 seconds total. The major design details of the two-stage hydrazine gas generator used during the Phase I testing are shown in Fig. 13. The major sections are the pilot, first and second stage.

The pilot stage consisted of an injector and stand-off manifold to spray hydrazine into a catalyst bed. The catalyst used for spontaneous decomposition was Shell 405 in a 20 to 30 mesh size. Several pilot bed configurations were employed during Phase I to optimize start times. The selected baseline bed had a diameter of 1.88 inch and a bed length of 0.25 inch. This baseline bed maximized the temperature of the pilot hydrazine decomposition products for the ignition source, and this bed was used for many of the Phase I tests and all of the Phase II tests.

The first stage was essentially a combustion chamber fed by the hot gas from the pilot section. Four spray nozzles introduced additional propellant into the stage. Two upstream nozzles are 0.029 inches in diameter and introduced approximately 20 percent of the flow near the ignition source. These nozzles were canted toward the downstream position. The remaining 80 percent of the first stage flow was injected radially through two 0.062 inch diameter nozzles at a downstream location. After test 179, these nozzles were changed to 0.037 and 0.073 inches in diameter, respectively, which had approximately one-half of the injection pressure drop. The combustor itself was 2.68 inches in diameter with an end plate containing a rounded entrance orifice which was varied from 0.450 to 0.750 inches in diameter during this test series.

The second stage, 6 inches in diameter, had 8 injection orifices. Four orifices were located in the diverging entrance end of the combustor. These spray nozzles were 0.077 inches in diameter and delivered 20 percent of the second stage flow. The remaining four orifices were downstream in the straight wall section and were .187 inches in diameter. Exit orifice is a rounded entrance hole 1.55 inches in diameter. In several test configurations a secondary catalyst bed was installed in the combustor immediately upstream of the orificed end plate.

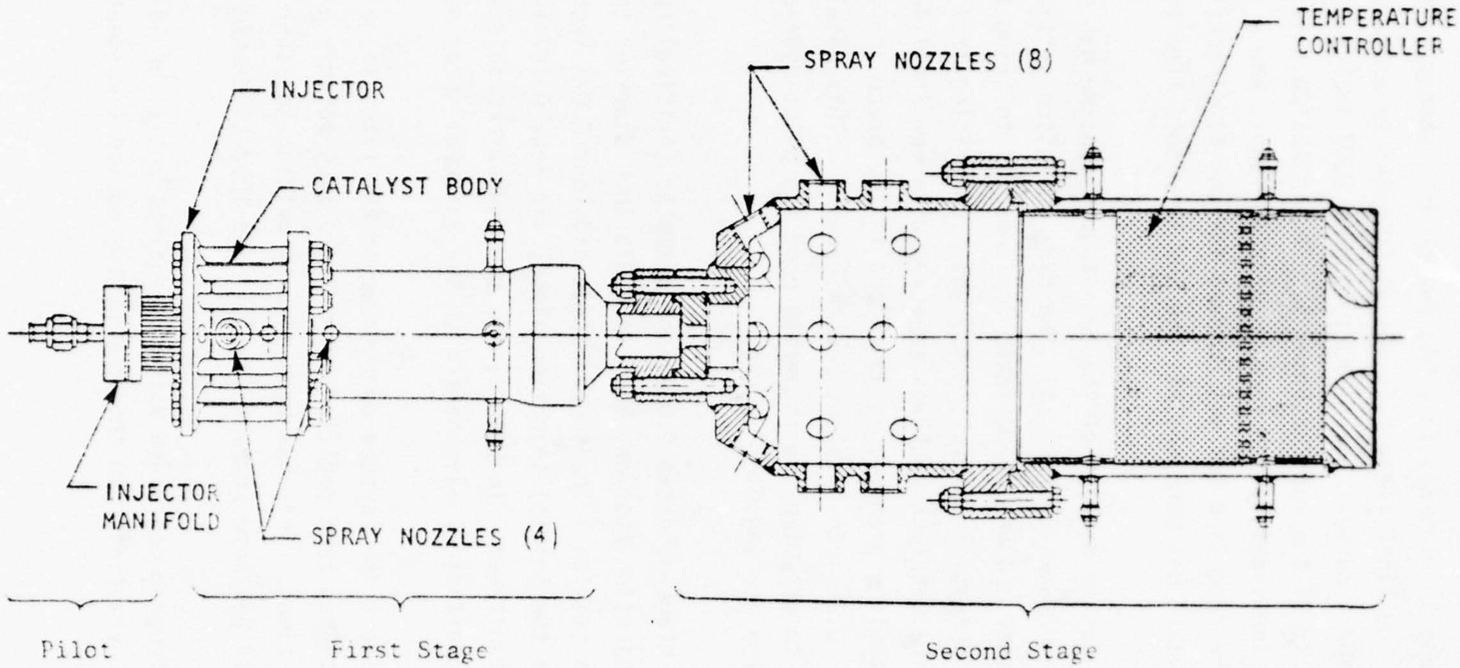


Figure 13 Brassboard Two-Stage Gas Generator Design

The pilot stage flow was controlled through an orifice which is varied from 0.029 inches in diameter to 0.043 inches in diameter in the various configurations. The first stage flow was distributed by branched lines to the spray nozzles, and flow was controlled through a 0.07 inch cavitating venturi for the majority of the runs in Phase I. The injection orifices of the second stage are individually fed through branced liners and a control orifice was used in the feed line but was removed for the final configuration 31. A separate valve was used to initiate and terminate flow to each stage.

The major design details of the two-stage hydrazine gas generator used in Phase II are shown in Fig. 14. The N_2H_4 injection section of the second stage reactor had been redesigned for Phase II to bring the propellant injection sprays closer to the hot gas flow exiting from the first stage nozzle. The catalyst pack was repacked with new Shell 405 catalyst and has the baseline geometry as developed in the preceding tests (1.88 inch diameter - 0.25 inch catalyst bed thickness). This catalyst pack was developed to maximize the temperature of the pilot hydrazine decomposition products for the ignition source.

The first stage used two 0.037 inch diameter (orifice) spray nozzles at the initial injection station, and two 0.073 inch diameter spray nozzles at the downstream station. The second stage had four 0.077 inch diameter spray nozzles in the initial injection plane, and four 0.187 inch diameter spray nozzles downstream. In both cases, the downstream injection spray nozzles were in longitudinal alignment with the upstream spray nozzles.

The propellant feed system downstream of the first stage and second stage main valves had been modified to minimize line volumes and thereby minimize line fill times. This had been done by welding together formed tubing sections to get more compact manifold and feed tube assemblies.

The test program proceeded without problems until the 141st test when severe propellant transients in the feed system caused a washout of the second stage.

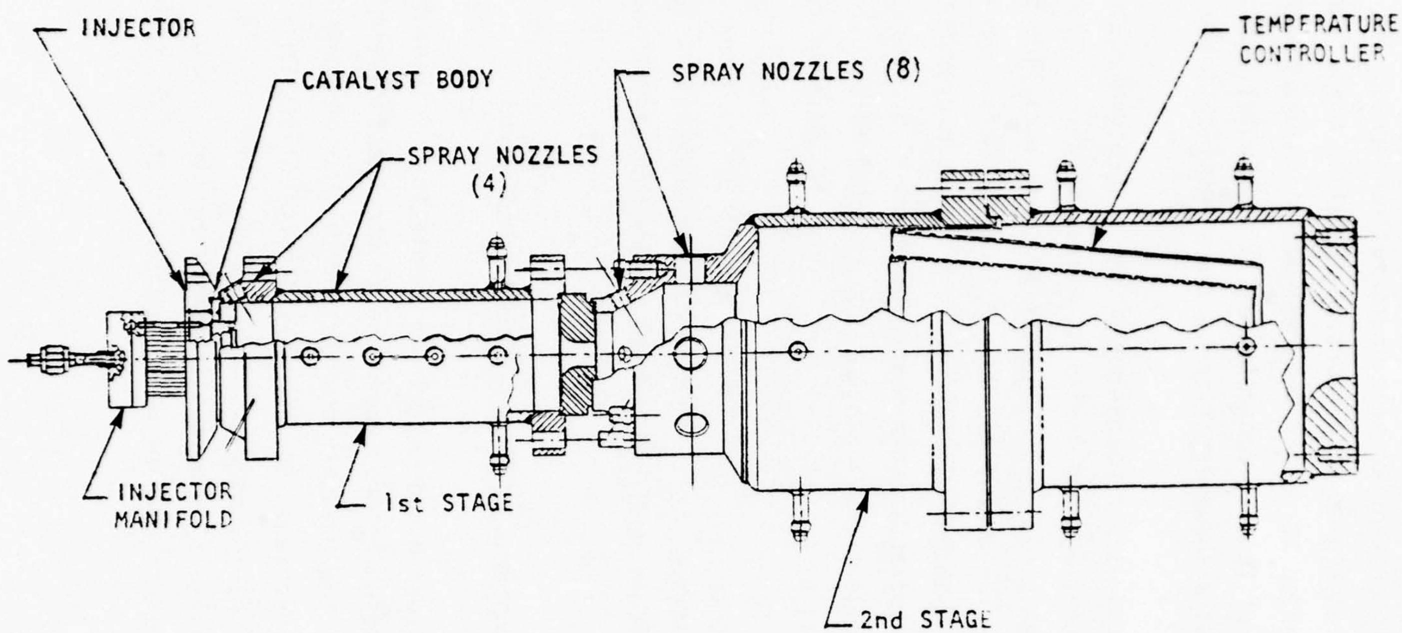


Figure 14. Modified Two-Stage Gas Generator Design

The propellant accumulated in the second stage detonated posttest, and the hardware was damaged. At this point in the program, all of the objectives had been completed. The gas generator rebuild incorporated feed system modifications to prevent adverse pressure transients, the second stage combustor was redesigned with improved operating margin, and sequence safety circuits were added to detect abnormal operation and provide safe shutdown. The last phase of testing consisted of 15 successful demonstrations of the gas generator with these facility/hardware modifications.

Smooth starts and stable operation of the staged combustion gas generator were demonstrated, and the limits of operation were explored. Two stage operation showed steady stage chamber pressure variation of only $\pm$ 0.5 percent. Operating conditions were defined to obtain smooth starts of the first and second stages with little or no chamber pressure overshoot. Stage ratio of the first and second stages of up to 11.5:1 were successfully demonstrated while overall stage ratio of total flow to the pilot flow of 91.4:1 were obtained. Various start sequences and feed system layouts were utilized, and it was found that the first stage could be started in less than 0.150 seconds even with the brassboard design hardware used. Test data shown that two-stage start times of 0.2 seconds are feasible through the use of flightweight manifold volumes, heated pilot beds and overlapping sequencing.

<u>Fast Start Demonstration (MOD. P00002)</u> [3, 4, 20]

Initial plans (which had called for completion of tradeoff studies and preliminary design before proceeding to component development) were modified as a result of the tradeoff studies and discussions during the first three months of the program. As more specific objectives were defined, a new task (MOD P00002) was added to the program, effective 14 June 1974, for immediate activation of a facility for demonstration of the fast start objective before proceeding with significant design and hardware modification.

An existing APU turbine previously used under contract F33615-715-C-1774 was provided and the brassboard gas generator unit developed under modification P00001 was integrated to conduct a three-phase program consisting of gas generator checkout, turbine simulation and fast start turbine inertia tests.

Test Series Description - The first test series reported herein was conducted without the thermal pack, and the fast start sequence was developed where turbine power was developed from 1 to 90 percent in less than 200 milliseconds. Next, the thermal pack was installed, and a slow start sequence was used to demonstrate the capability for temperature control. Fast starts could not be obtained with the thermal pack in place, indicating that the presence of the thermal pack interrupted the combustion chamber geometry required for the fast start mechanism. Therefore, the thermal pack was relocated into a downstream plenum. The staged combustion gas generator and downstream plenum for the thermal control pack were integrated with a hot gas bypass safety valve, installed to provide an additional method of terminating power to the turbine. A turbine simulation test series was initially conducted without the turbine using a duct and orifice to simulate the turbine inlet manifold and nozzles. This series also verified operation of the new generator setup with the throat located downstream of the combustion chambers.

Finally, the test facility was prepared for the fast start turbine inertia testing using the existing APU turbine, a 3.2:1 reduction gearbox, and the inertia simulator. A series of start tests was conducted with the 4.0 and 2.88 slug-ft^2 flywheel assemblies and with gas power levels of approximately 100 and 105 percent. Starts in less than 1 second were demonstrated. During the three series of tests, 60 hot fire tests were conducted, and data was obtained on the start characteristics of the turbine assembly. Additional data was also obtained on the operating characteristics of the staged combustion gas generator including rapid start sequencing and catalytic thermal control.

Test Results - The start characteristics were established for the Mark 15-EO Model turbine which simulated system inertia similar to the prototype turbine for an airborne APU. The turbine was driven by the exhaust products of a staged combustion gas generator of the type to be used on the APU system, and additional information was obtained on the gas generator operation.

The transient turbine tests were run during which the turbine was accelerated to a speed near the safe operating speed and then power was cut off. Pressure and flowrate were relatively constant during the start transient as plotted in Fig. 15. Temperature is much more slow to rise and does not reach steady-state because the temperature probe, a thermocouple ungrounded to a tube, has a significant thermal lag. The fact that pressure and flow were essentially constant implies that temperature was also, except for the small amount of heat going into the gas generator and ducting walls. Turbine data reduction and analysis concluded that temperature ran at about 1650°F.

Gas generator chamber pressure buildup from 1 to 90 percent of chamber was achieved in less than 200 milliseconds as shown in Fig. 16, using the existing facility type valves and manifold volumes. The pilot was started and conservatively operated for two seconds; however, the gas flow (1%) was insufficient to spin the turbine and did not affect the objectives of the fast start turbine.

Figure 17 describes the predicted acceleration curve based on total system inertia - which includes inertia of the turbine, the gearbox and the flywheel rotor. The points at 0.482 slug-ft^2 simulate operation with a 0.281 slug-ft^2 alternator and the points at 0.592 slug-ft^2 simulate operation with a 0.391 slug-ft^2 alternator.

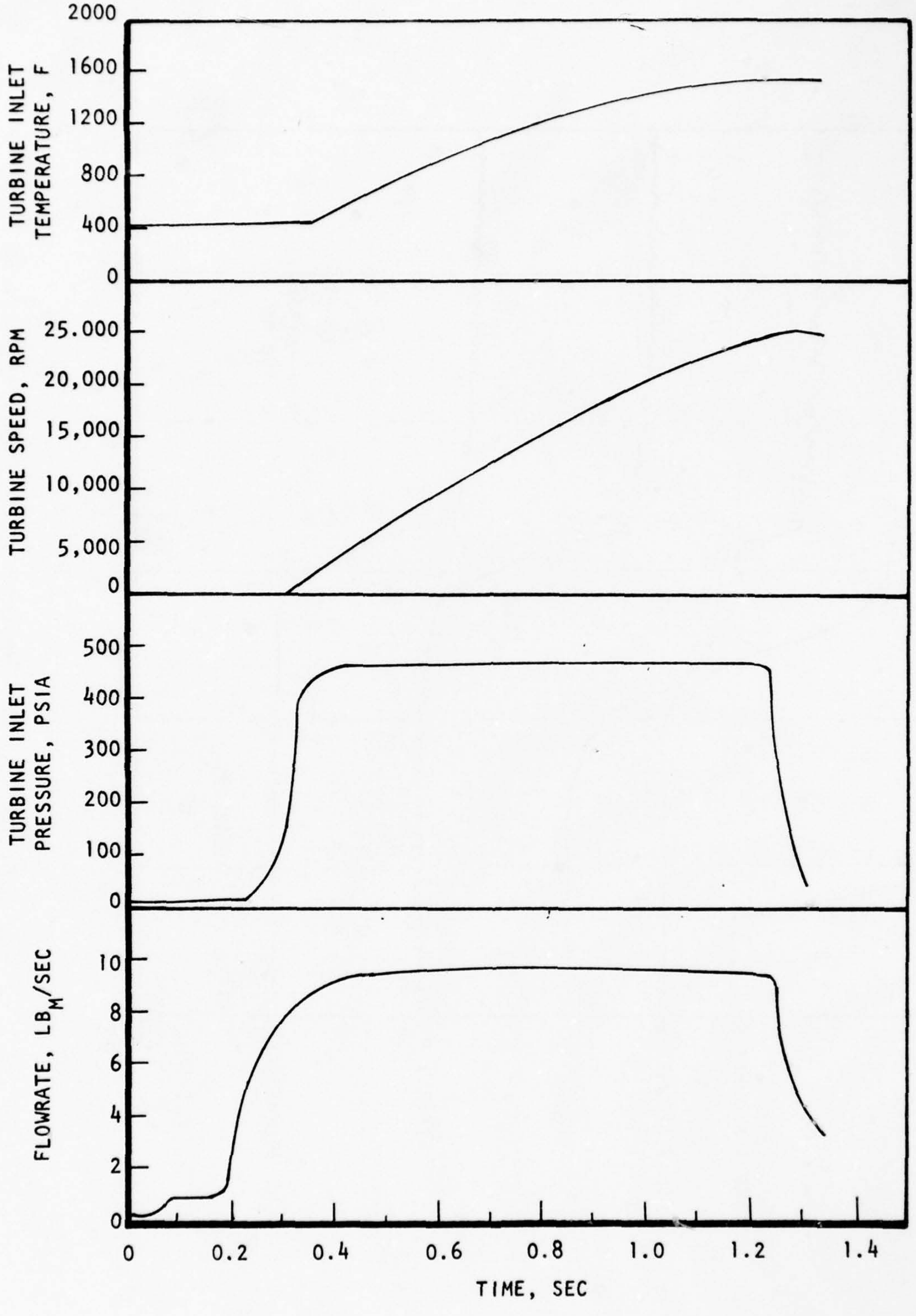


Figure 15. Test 59 Data Effective Inertia 0.281 Slug-Ft^2

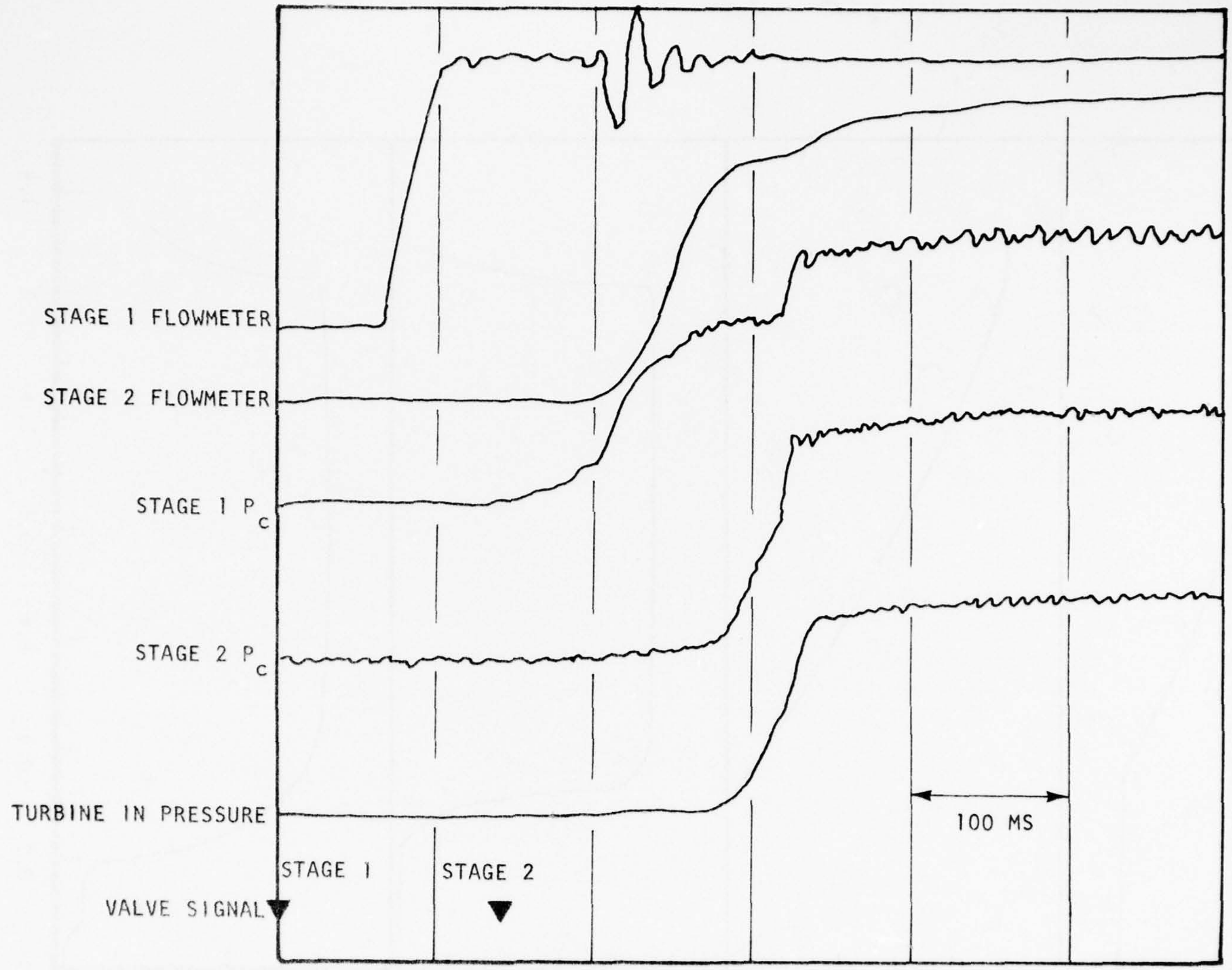


Figure 16. Gas Generator Start Transient, Test 59

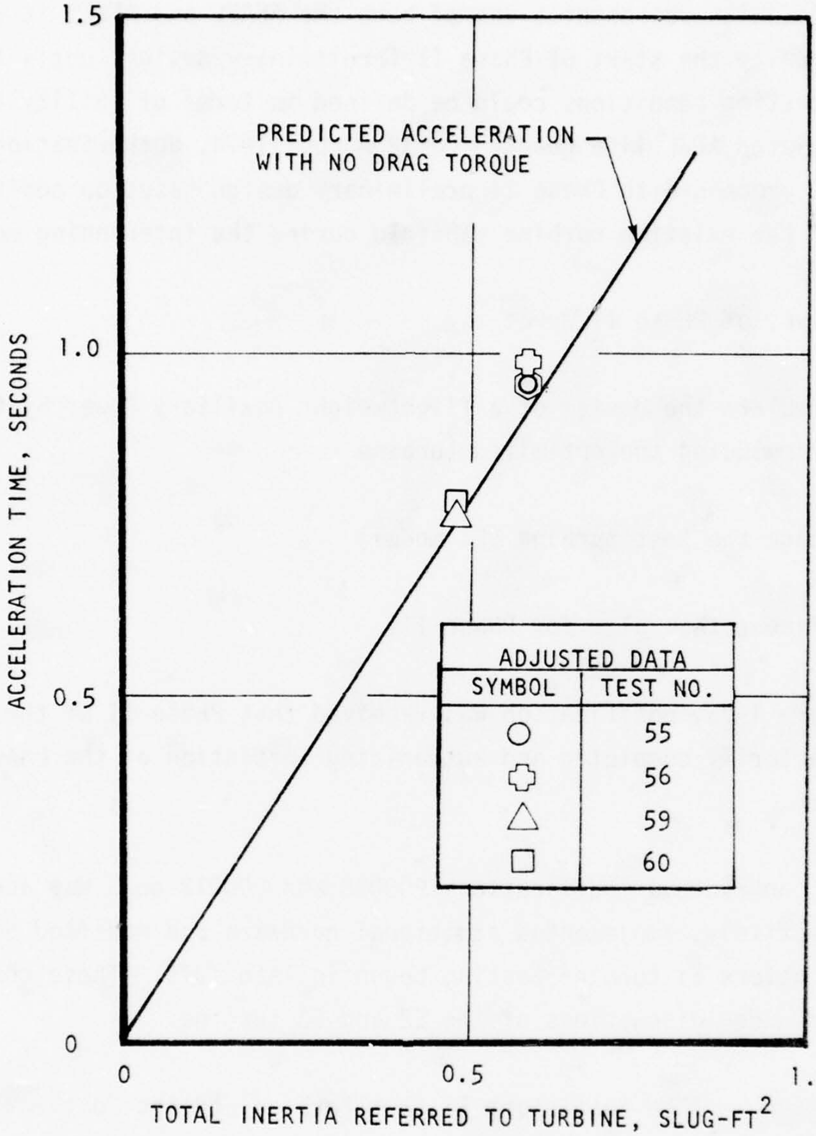


Figure 17. Acceleration Time Comparison

2. PHASE II

Preliminary Design (E1) [5, 6]

Introduction - Following review of the Phase I (tradeoff study) on 12 June 1974, with representatives of both the AFAPL and AFWL, it was decided to delay the start of Phase II (preliminary design) until test turbine operating conditions could be defined in terms of ability to meet newly formulated AFWL life needs. On 12 August 1974, authorization was received to proceed with Phase II preliminary design based on additional analysis of the existing turbine manifold during the intervening months.

The objectives of Phase II were:

(a) To formulate the design of a flightweight Auxiliary Power System (APS) including the optimized turbine

(b) To design the test turbine (E1 model)

(c) To define a test plan for Phase III

On 21 January 1975, notification was received that Phase II of the program was satisfactorily completed and authorizing initiation of the Phase III test program.

Subsequent contractual modifications P00008 and P00012 on 5 May and 25 Sept. 1975, respectively, implemented additional hardware and modified design parameters before E1 turbine testing began in late 1975. These changes are covered under discussions of the E2 and E3 turbine.

Turbine Design - The fast start E1 test turbine and the "optimum" turbine were designed to meet the conditions of Table 8. Subsequently, these criterian were revised to specify requirements for an E2 model deliverable turbine. The horsepower and specific propellant consumption were part of

the original contract requirements. The operating conditions of Table 8 were the result of the systems study of Phase I of the contract. This study, selected the best set of operating conditions (primarily on the basis of system weight). The inlet temperature, for example, represents the highest which could readily be used without exceeding turbine stress limitations. The turbine rotative speed represents the highest that could be used with some additional stress margin and essentially no sacrifice in weight. Inlet pressure represents a value slightly higher than the optimum but one which produces a better stage Mach number distribution. The turbine exhaust pressure represents the lowest value that can reasonably be obtained at sea level with a practical exhaust duct length.

The design meeting the requirements of Table 8 is also subject to some design constraints. A major constraint for economy purposes is that the E1 test turbine should be able to use the turbine inlet manifold and the bearing and shaft arrangement from the earlier Rocketdyne E0 turbine. Thus, the turbine diameter is fixed within a very narrow range. In order to satisfy various experimental needs of the Air Force, the life of the turbine should be at least 100 simulated missions at an inlet temperature of 1450°F, following approximately 30 simulated missions at 1650°F temperature. The lower temperature associated with the hundred missions is primarily to allow this life to be achieved with the existing inlet manifold.

TABLE 8. E1 DESIGN CONDITIONS

Hydrazine - 62% dissociated	
Inlet temperature	2110°R (1650°F)
Inlet pressure	700 psia
Exit pressure	17 psia
Rotative speed	29,000 rpm
Power	6000 hp
Propellant consumption	4.5 lb/hp-hr

Desired life of the test turbine must be met while satisfying safety factor criteria essentially the same as those applied by Rocketdyne to the design of the reuseable Space Shuttle Main Engine. An additional constraint on the design is the acceleration time. The turbine shall have a low enough inertia and high enough torque to accelerate a load inertia equivalent to 2 slug-ft^2 at 8000 rpm to full speed in 0.85 seconds. (This allows 0.15 seconds for gas generator starting so that the whole start sequence will take place in less than one second).

An optimum turbine would have a new inlet manifold specifically designed for this application. Both turbines would have the same aerodynamics except for the inlet manifold flow passage. Designs for both turbines were produced in Phase II of this contract. However, only the test turbine was fabricated and tested in Phase III.

The next section of this report describes the techniques and some of the key results of the various kinds of analysis performed in the design of the turbine. The following section presents the actual mechanical design and describes some of its features. The final section describes the optimum flightweight gas generator design concept.

Aerodynamic Design - The aerodynamic design of the first stage of the turbine is supersonic and the second stage is subsonic. That is, the flow entering the rotor of the first stage is supersonic while the flow entering the rotor of the second stage is subsonic.

The potential flow characteristics of the first stage nozzle were calculated using two Rocketdyne computer programs. The first of these calculates the subsonic and transonic flow in the throat region of the converging-diverging nozzle. The second uses the information from the throat calculation to define the walls of the diverging supersonic portion of the nozzle utilizing the method of characteristics. The two programs thus allow definition of a nozzle which will produce uniform, parallel, supersonic flow at the exit plane of the diverging portion of the nozzle. Since these are basically rocket nozzle design programs, they must be laid out with the nozzle centerline at an angle to the plane of the wheel in order to produce a tangential momentum. With this sort of arrangement, one wall of the nozzle must be extended parallel to the centerline of the nozzle.

With the potential flow boundaries of the nozzle defined, the boundary layers on both walls were calculated utilizing a boundary layer computer program. The physical walls were moved back from potential flow surface by the displacement thickness of these boundary layers. The trailing edge thickness was maintained at a minimum value at the hub. It should be noted that the nozzles as designed are two-dimensional. Basically, this means that two walls are contoured and two walls are flat. Centerlines of the nozzles are straight lines. These nozzles were made in this fashion rather than with the more common approach of nozzles wrapped around a cylinder. Previous Rocketdyne experience with nozzles designed for two-dimensional supersonic flow, then wrapped into a three-dimensional form, proved that the wrapping process introduced significant additional losses into the flow from the nozzles. The additional losses associated with thick trailing edges associated with straight nozzle centerlines are much less than those developed in the wraparound configuration.

An important aspect of the rotor design is the selection of the appropriate inlet relative velocity. In subsonic turbines, typically, the absolute velocity leaving the nozzle is calculated assuming that the wakes associated with the surface boundary layers and the trailing edge thicknesses of the blades are mixed completely with the core or potential flow through the blades. In the subsonic case, with relatively thin trailing edges and thin boundary layers this is a valid assumption. The supersonic case is somewhat different. The wake mixes at a relatively slow rate in the supersonic case. Hence, the rotor blades for most of the time are exposed - not to a mixed flow - but to the core flow of the nozzles. For a short period of time each blade passage is exposed to the portion of the flow containing the nozzle wake.

Design of the supersonic rotor to accept the core flow velocity rather than a mixed fluid velocity complicates the assessment of rotor losses. The approach taken was to examine the momentum change through the rotor rather than the energy losses. The key assumption in the analysis, analogous to that successfully used in the partial admission turbine, is that the fluid has completely mixed and is homogenous by the time it leaves the rotor.

Losses in both the rotor and stator were basically calculated using the potential flow surface velocity distributions, boundary layer momentum thickness, and profile loss calculations. Nozzle end wall losses were calculated utilizing the same boundary layer data. Rotor end wall losses and rotor clearance losses were calculated using Balje relationships.

Combining the potential flow and boundary layer calculations for the rotor required careful consideration. The inlet condition to the rotor was calculated with the finite thickness leading edge accounted for by turning and acceleration of the flow such that continuity is satisfied before and after the turning necessitated by the leading edge thickness. The trailing edge is somewhat more complex. With the leading edge thickness fixed, a calculation of the potential flow through the rotor for a fixed exit angle produces a particular potential flow trailing edge thickness. Calculation of the boundary layer displacement thickness and subtraction of this from the potential flow surface defines the physical trailing edge thickness. Iteration on the exit angle is required to find that exit angle which results in exactly the desired trailing edge thickness.

Detailed design calculations during the early phases of the first stage design showed that the selected pressure ratio for the first stage produced a rotor blade height which was quite small. Tip clearance and end wall losses were thus very significant. Calculation of the effect of increasing the stage pressure ratio, that is, decreasing the first stage exhaust pressure, showed that performance could be improved. While the increased pressure ratio results in increased isentropic velocities, thus a lower turbine velocity ratio, the increased blade height due to expansion to the lower pressure level reduces the end wall and clearance losses more than enough to make up for that. The first stage nozzle design pressure ratio was thus increased as compared to the original valve.

The first stage performance is still very sensitive to radial tip clearance even with the higher blade height. Because of the sensitivity, considerable effort was devoted to controlling the clearance.

The use of momentum analysis for calculation of power produced by the supersonic rotor has one significant disadvantage. It does not allow the direct calculation of the pressure at the exit of the rotor. The approach which was taken was to make the assumption that the absolute flow downstream of the rotor which is subsonic was completely mixed at the entrance plane to the second stage stator. Continuity must be satisfied at that location.

The absolute velocity leaving the wheel is determined from the momentum calculations, while the absolute total temperature is determined from the energy extraction. These, combined with the flowrate and the area, allow the computation of the flow Mach number, hence, the corresponding total pressure. With total pressure defined, the static pressure can also be calculated. This approach then defines the after-mixing conditions which constitute the inlet conditions for the second stage stator.

The second stage is basically a subsonic stage having a pressure ratio across the nozzle slightly greater than the critical. Because of the larger blade height associated with the exhaust pressure of 17 psi, the second stage could no longer be considered on a single streamline basis. The design has, therefore, been made as a free vortex design. Normally, the use of a non-free or controlled vortex swirl distribution would be expected to produce better performance than free vortex. In this case, however, calculations have shown that the potential improvement is insignificant.

Potential flow calculations in the second stage stator and rotor were made initially using the Douglas-Neuman technique. Final refined calculations were made using TSONIC, NASA's transonic cascade calculation program. Calculations were made at the hub, mean, and tip radial locations. Geometry of the blade at the mean was generated from straight line segments joining the hub and tip profiles.

Losses for the second stage were evaluated in the same manner as for the first stage. The assumption of complete mixing of the wake was used in the case of the subsonic machine, however. Loss calculations were performed only at the mean streamline.

The predicted specific propellant consumption of 4.5343 is within 1% of the desired value of 4.50. The predicted efficiency is just under 70%. It should be noted that the reduced flow in the second stage results from labyrinth seal leakage between the two stages. This flow is assumed to be ineffective in the second stage.

Thermal Analysis - A series of thermal analyses was also conducted during the design. These had two primary purposes: to provide data for stress calculations and to provide aerodynamic data, such as tip clearance and throat area variations. The stress calculations involve transient and steady-state thermal gradients as well as steady (or near steady) state temperature levels in highly stressed parts.

Most of the design analysis centered on the turbine nozzle and rotor blades, rotor disks, and the tip shroud structure of both stages.

E1 Test Turbine Design Description - The assembly of the E1 test turbine is described by drawing No. XEOR 941470. The basic turbine manifold and rotating assembly are those previously used on the Air Force sponsored (Contract F33615-71-C-1774) Airborne Power Unit. The turbine wheels, stators, tip seals and instrumentation are different.

One significant difference between the E1 turbine assembly and the previous E0 turbine is in the method of attaching and locating the turbine wheels. The E0 turbine utilized a stud drive which relied on precise radial location of studs and precise diametral control of them for piloting of the two turbine wheels. The E1 design utilizes curvic couplings to center the wheels and less precise studs to clamp the wheels axially. A brief design study showed that the curvic coupling approach would be simpler and more economical. The curvic design is made even more economical by utilizing existing tooling and existing studs. Use of existing studs explains the apparent excess material on the second stage wheel next to the locks and nuts.

Turbine wheel materials considered were Astroloy and thermal-mechanically-processed (TMP) Waspaloy. Properties are almost identical for the two materials. The Waspaloy was chosen on the basis of availability and of Rocketdyne experience in use of the material for the Space Shuttle Main Engine.

Another design study was performed to determine the best abradeable material for use as a turbine rotor tip seal material. Three alternate materials were studied. Wall Colmonoy's Nicroseal, Union Carbide's Ucar Type AB-1, and Brunswick's Feltmetal all appear to offer similar, technically acceptable characteristics. The Feltmetal seal was initially selected on the basis of minimum engineering and fabrication effort being required. Nicroseal would require a Rocketdyne design worked out with experts in Detroit with the seal material being applied to a Rocketdyne backing ring in Detroit. UCAR Type AB-1 also required a Rocketdyne design worked out with experts in Cleveland with seal material being brazed onto a backing ring by Rocketdyne. With the Feltmetal approach, a local representative does the design liaison and the seal can be procured as a completed assembly. Cost and schedule considerations determined the selection of UCAR Type AB-1.

The interstage seal between the first and second stage is a labyrinth type seal which also uses the same abradeable material. The sealing surface was selected as the smallest diameter which could be used, the outer diameter of the curvic coupling.

The detailed design of the first stage turbine tip seal was selected to match as closely as feasible the radial growth of the blade tips throughout the period of operation. The factor which makes this difficult is that the tip seal, being directly exposed to the hot gas, heats up quite rapidly. If the tip seal were a complete hoop it would therefore expand away from the rotor rapidly. The rotor, on the other hand, has considerable thermal mass. Thus, a significant period of time is required before thermal equilibrium in the rotor is approached. The control of the seal growth is achieved by segmenting the hoop and supporting each segment on a radial pedestal. The pedestals in turn are attached to a larger diameter hoop whose thermal characteristics thereby become the controlling element.

The second stage tip seal is fabricated as a complete hoop. This seal is subjected to significantly lower temperatures than the first stage. The second stage rotor blades which heat up rapidly are considerably longer, thus allowing the rotor tip to follow the seal more closely.

The bearings shown in the E1 test turbine assembly are those previously used in the E0 unit.

Some additional features peculiar to the test turbine should be pointed out. The first stage turbine nozzle is welded to the existing manifold. The cross sectional area of the turbine nozzle structure is such that it is not sufficient to withstand the radial pressure forces attempting to separate the nozzle inner and outer walls. Two approaches to this are possible. The inlet portion of the nozzle can be modified to thicken the walls in that region, thereby giving an effective inlet angle of the nozzle closer to axial. This approach must be used in the optimum machine. The other approach which was adopted to the E1 test turbine is to utilize the existing nozzle structure to withstand most of the separating forces. This approach was adopted at a time when it appeared that the first stage nozzle would be made from Hastalloy C276, a relatively weak alloy. Although the nozzle material was later changed to the much stronger Rene' 41, the design was not changed.

The thermal insert shown in the turbine manifold inlet region was designed to prolong the life of the manifold. As determined earlier in the heat transfer analysis such an insert significantly reduces the thermal gradient to which the thick inlet flange is subjected. For convenience, the thermal insert was attached to a new flanged insert which also holds the inlet instrumentation. The critical speed of the test turbine rotating assembly was calculated as a function of bearing stiffness. The first critical speed is below 11,000 rpm for all reasonable bearing stiffnesses. Second critical speeds were calculated to be above 38,000 rpm for the same range of stiffness. Thus, the operating speed of 29,000 rpm represents a speed between the first and second critical with a margin of more than 20% between the operating speed and closest critical.

Optimum Flightweight Gas Generator Design

A preliminary flightweight design of the full-size gas generator was produced in September 1974 as shown in Fig. 18 Nozzle arrangement and empty chamber volumes and shape are the result of the successful test configuration of the gas generator test program. The gas generator produces 8.48 lb_m/sec. of hot gas delivered to the turbine at 1560°F and the nominal flow distribution is 0.1 lb_m/sec. through the pilot, 0.75 lb_m/sec. into the first stage and 7.63 lb_m/sec. into the second stage. Total weight of the assembly is 32 lb_m with a displaced volume of 0.37 cubic feet.

The use of very low flowrates in the catalyst pilot permits design of the catalyst pack for long life by using a bed loading of 0.02 to 0.03 lb_m/in^2/sec. A life correlation for Shell 405 catalyst shows that this conservative bed loading should give a lifetime exceeding 6.7 hours. The bed utilizes the design optimized during the test program. The pack is 1.88-inch diameter by 0.23-inch long and contains 14 grams of Shell 405 catalyst which has a value of approximately $100. The bed temperature is maintained above 80°F by electrical heat requiring approximately 20 watts of power, and a bolted joint is used for ease of inspection or replacement of the pilot bed.

The stage injection schedule is similar to that used on the tested gas generator. Four injection nozzles are used in the first stage where two nozzles inject 20 percent of the stage flowrate just downstream of the pilot outlet, and the remaining 80 percent of the flow is injected at a downstream location. This same internal staging schedule is used in the second stage except four nozzles are used at each location. Injection pressure drop for all nozzles is 250 psid.

Exhaust gas temperature control is achieved through the use of an ammonia decomposition catalyst in the gas phase below the reactor. A bed of the commercial ruthenium catalyst Shell X-2 and X-4 will reduce the output temperature to 1650°F with less than 3 lb_m of the low cost catalyst. Bed pressure drop would be approximately 20 psid.

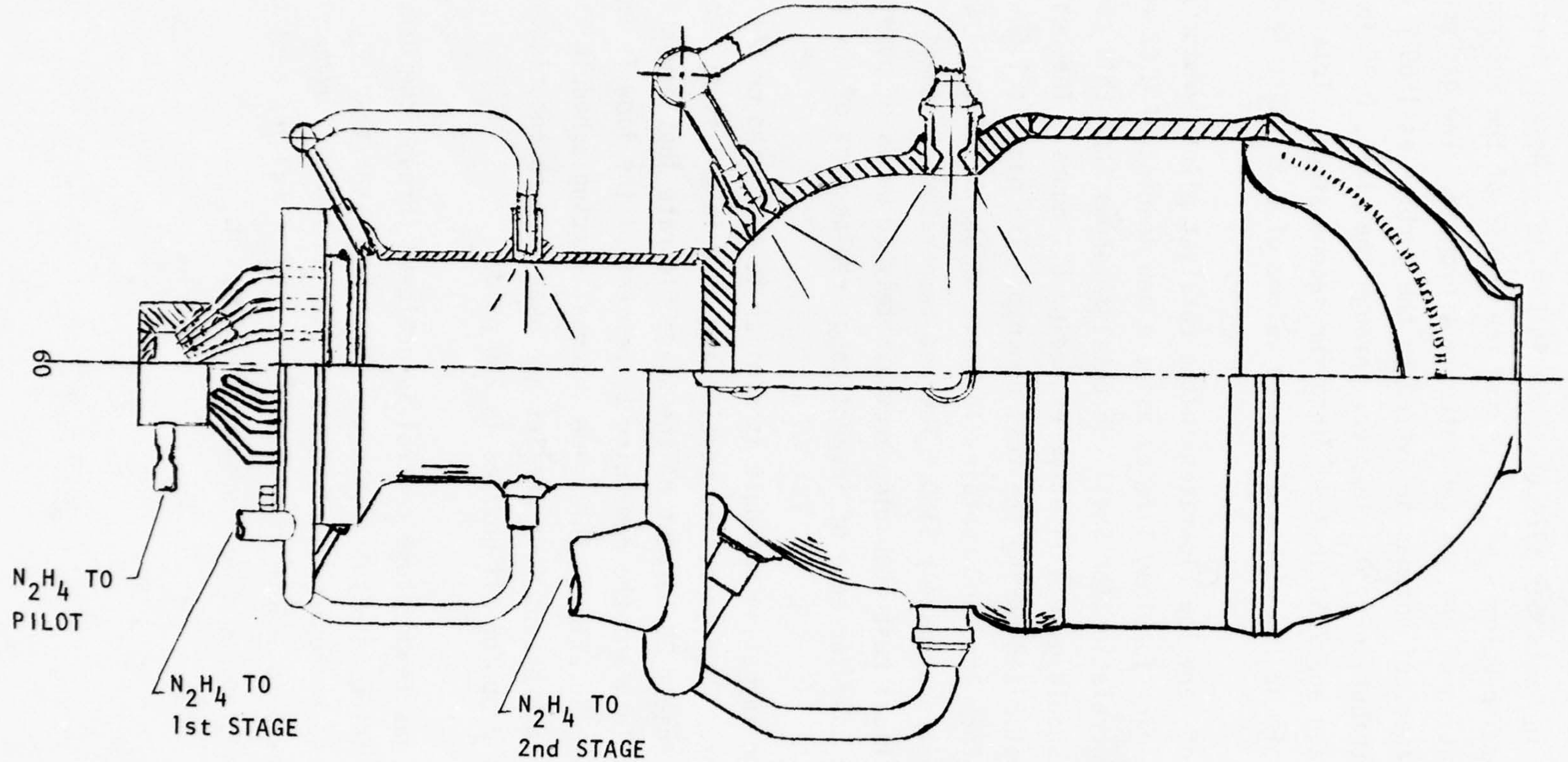


Figure 18. Preliminary Flightweight Gas Generator Design

The primary material for construction of the gas generator has been selected on the basis of nitriding resistance as well as high strength to contain potential overpressures. All structural components will be fabricated from Inconel 617 which has demonstrated the best nitriding resistance of high strength, high temperature materials.

Safety monitoring circuits similar to those used during the test program would monitor the gas generator operation. If chamber pressure does not rise as expected or falls below a minimum level during the operation, the gas generator would be rapidly shut down. Calculations show that this action would be completed to prevent the accumulation of more than 0.19 lb_m of liquid hydrazine. In the worst case, if all this liquid detonated, stress levels in the pressure shell would be below 1/3 of ultimate, resulting in no damage.

3. PHASE III

El Turbine Performance Tests [7, 8, 9]

Introduction - During the period of March 1975 through January 1976, a Mark 15-El turbine was produced and tested for evaluation of its performance. This assembly was the first turbine specifically bladed for the fast start program performance objectives, however, most other components of the turbine were as used originally in the J-2 rocket engine turbopump.

This higher pressure ratio El design maintains a supersonic flow condition from the throat section of the first stage inlet nozzles, through the first stage rotor blade passage channels. In addition, the incorporation of labyrinth type hot gas rim seals in conjunction with new aerodynamics substantially reduces (but does not eliminate) the unbalanced thrust load on the turbine discs. The hydrostatic thrust bearing oil delivery requirement was reduced to approximately 10 gpm of MIL-L-7808 oil at 100 psia. The ball bearing nearest the wheels was also lubricated from this source. The outboard ball bearing was lubricated by a jet in the gearbox.

In order to define the performance of each turbine stage, internal porting was provided to allow measurements of the interstage pressures. These capabilities did not exist in the Mark 15-EO turbine.

The primary objective of this experimental program was to evaluate the performance of the Mark 15-El turbine over the complete range of anticipated use conditions with two different turbine inlet temperatures. These tests required the use of a dynamometer to continuously absorb the turbine developed power, while maintaining a desired turbine speed.

Test Program - The initial seven (7) tests of this series were performed using only the hydrazine gas generator portion of the system. The gas generator exhaust products were ducted directly into the facility exhaust system using a 1.394-inch diameter orifice to simulate the turbine nozzle throat area. The initial four firings were for durations of 5 seconds or less.

The main objective of these tests was to check out the stand operation and particularly, the ability of the facility exhaust system to handle long duration firings. The final three firings were for 10 seconds duration at propellant flow conditions which would be expected to develop 50 percent, 75 percent and 100 percent of the design turbine horsepower.

Next, the Mark 15-E0 turbine which had been used for fast start turbine demonstration was installed for initial checkouts of the dynamometer system prior to availability of the Mark 15-E1 turbine assembly. A series of 17 exploratory tests were conducted, 10 of which were full duration of 10 seconds. During these tests, the dynamometer control system was the primary item of interest, with the major effort devoted to establishing procedures which would allow acquisition of steady-state power data in the 10 second duration runs. The major problem encountered was that the dynamometer has a tendency to unload during the initial start transient, resulting in an overspeed condition, terminating the run. Addition of an inlet pressure feedback control loop considerably increased the controllability of the system, however, it was still found necessary to anticipate this tendency for the turbine to overspeed and to manually override the inlet valve control near full speed. When the turbine speed leveled off, it was then necessary to adjust either the inlet or outlet valve setting to attain the target turbine speed level.

The startup procedure which was adopted programmed an initial period of approximately 1.5 seconds of operation with only the first stage of the gas generator active. This resulted in an approximate 6,400 rpm maximum possible turbine speed. This portion of the initial acceleration, therefore, was not subject to any possible overspeeding, and allowed the test operator about 1 second of time during which any desired valve manipulations could be initiated. Additionally, this first stage only operation resulted in a certain amount of turbine preheating, since inlet temperatures did not exceed the 500-700°F range. This considerably reduced the temperature shock on the turbine structures which are directly contacted by the hot gases.

An additional system modification instituted for these tests was the revision of the turbine lubrication system. The thrust bearing of the Mark 15-EO turbine had shown a tendency toward a high wear rate and metallic transfer between the steel and bronze wear surface. Internal turbine pressure measurements obtained resulted in the calculation of considerably high thrust loads than had been previously assumed. This indicated that a higher inlet oil pressure was required for this hydrostatic bearing. Accordingly, these tests used a 1450 psig turbine lubrication system oil pressure, as opposed to the 700-900 psig range previously employed. Posttest inspection of this thrust bearing showed no evidence of any significant bearing surface attrition. Finally, a series of 23 firings were performed with the Mark 15-E1 turbine installation.

The final series of firings were performed using the same operational procedures previously developed. However, since analysis of the internal pressure data for the turbine showed that the thrust load acting on the hydrostatic bearing was much lower for the Mark 15-E1 design as compared to the Mark 15-EO, the turbine inlet oil pressure was reduced. It was intended to reduce this pressure from the former 1450 psig level to 300 psig. The high pressure pump system could not be successfully regulated to such a low value, and tended to creep higher with time. Consequently, the actual thrust bearing lubrication oil pressure ranged from 420 psig to 670 psig for the various tests, and was reasonably constant at some value within this range during the time of the actual test. Use of a higher lubrication oil pressure than actually required to counter the thrust load is undesirable since losses associated with oil churning are very significant.

Results

1. The Mark 15-E1 supersonic turbine demonstrated a Specific Propellant Consumption (SPC) at rated power of 4.5 to 4.6 lb/hp-hr, as compared to a goal to 4.5 and a maximum of 4.77 (Ref. Fig.19).

2. Axial thrust of the Mark 15-E1 turbine is greatly reduced as compared to the Mark 15-EO. Oil flow above 6 lb/sec to the thrust bearing tends to flood the bearing cavity, resulting in an unwarranted power loss.

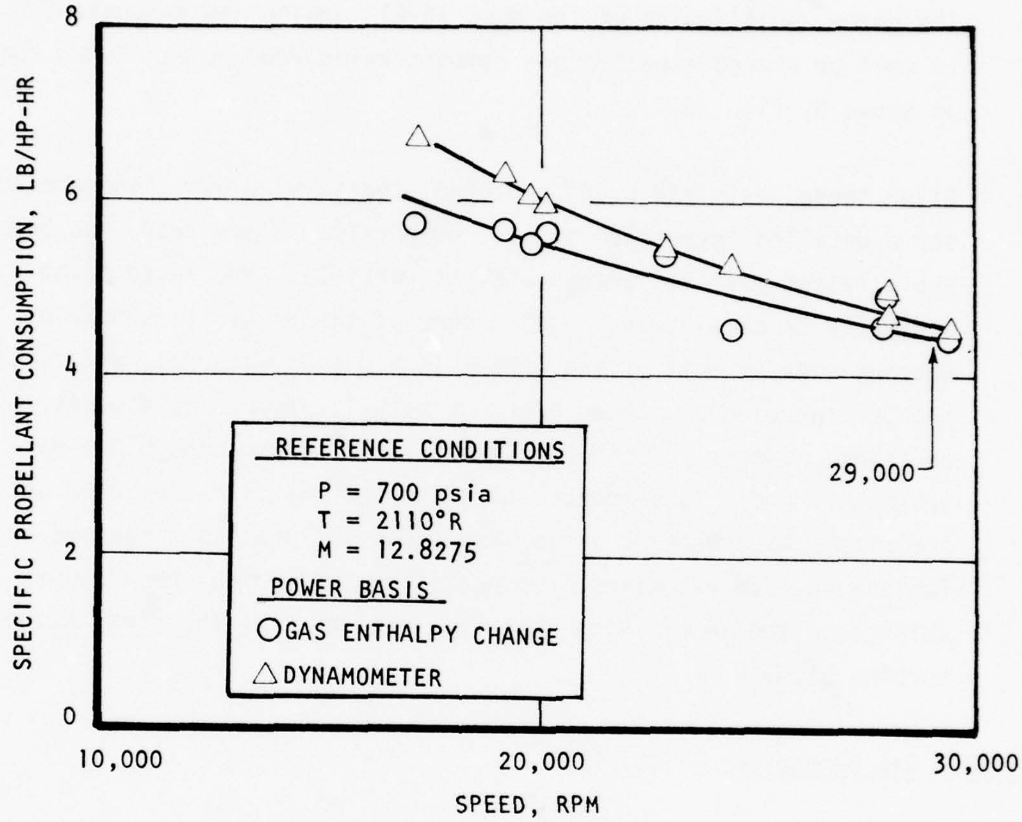


Figure 19. E1 Turbine SPC

The demonstrated axial thrust is sufficiently low such that consideration may be given to future elimination of the hydrostatic thrust bearing, the first stage rim seal, or both.

3. The dynamometer gave accurate and consistent results.

4. The aerodynamic design of the Mark 15-El turbine was demonstrated to meet or exceed expectations demonstrating 6000 hp at 29,000 rpm, as shown by Fig. 20.

5. After these tests had been completed, the turbine was disassembled for a detailed inspection of all components. Since there had been two catalyst pack failures, catalyst particles were noted to be partially or completely plugging some of the first stage nozzles, wearing away of most of the abradable tip seal material, and coating the turbine blades with an adherent metallic-appearing deposit. In addition, however, a corner of one first stage turbine blade had broken off and a large number of blade root cracks were noted on the first stage wheels. This mechanical failure was diagnosed to be fatigue failure resulting from operation in turbine speed ranges which resulted in exciting a natural frequency of the first stage turbine blades.

1450°F Gas Generator [10, 11]

This task was initiated in compliance with contract MOD P00016, effective 26 April 1976 to:

1. Demonstrate the ability to produce a 1450°F output temperature at design hydrazine flowrates (8.33 lb/sec).

2. Demonstrate the integrity of the catalyst pack over a series of 50, ten-second duration firings.

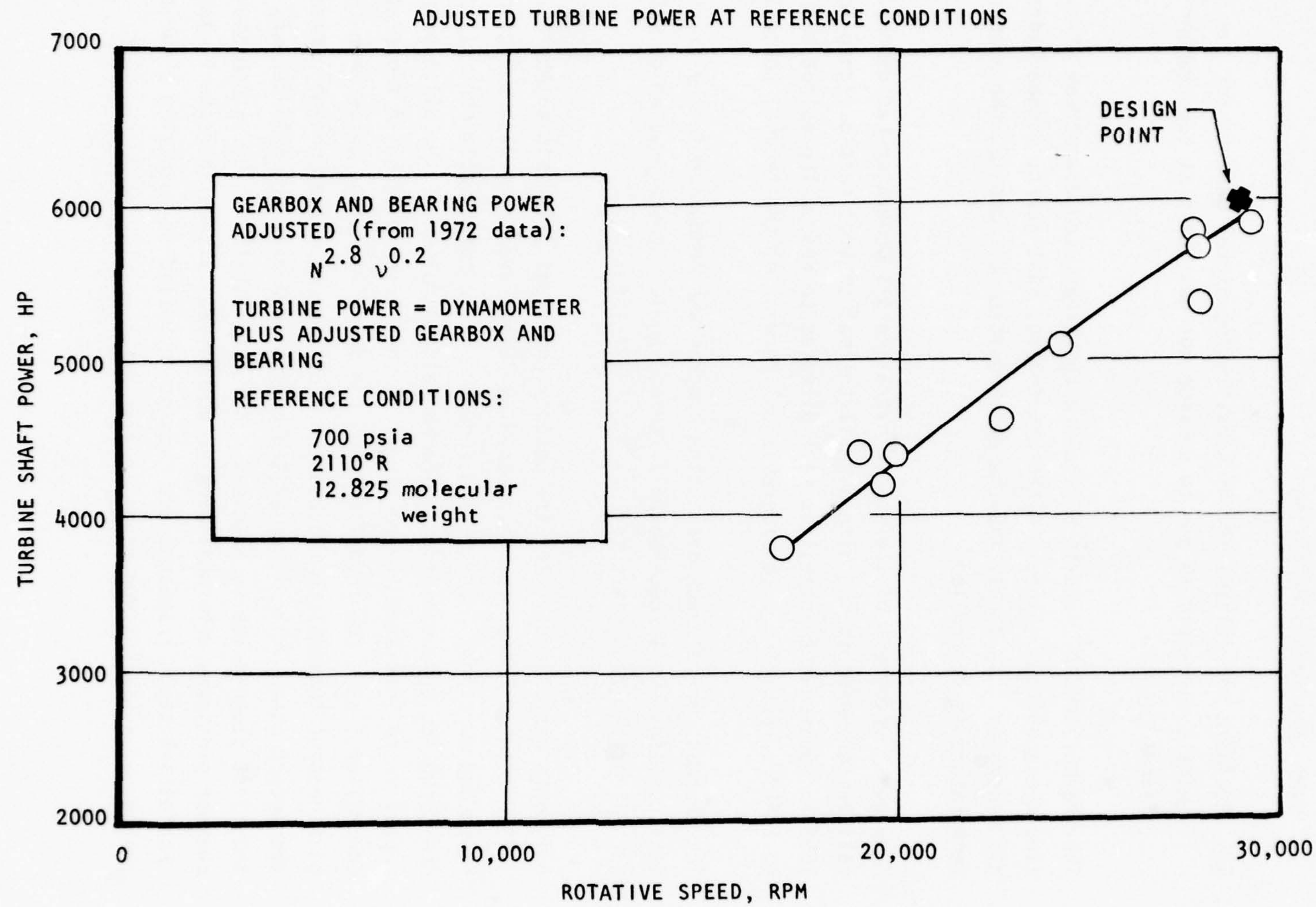


Figure 20. E1 Turbine Power

3. Determine the loss rate of catalyst fines, which will be generated by mechanical and thermal forces acting on the catalyst particles, over this series of firings.

4. Evaluate the suitability of substituting nickel gaskets for the copper gaskets currently in use to provide hot gas seals at the chamber flange joints.

The thermal control pack is a catalyst container used downstream of hydrazine decomposition volumes in the breadboard, fast start, staged hydrazine gas generator. By catalyzing the decomposition of ammonia, the output gas temperature is controlled.

Previous development of the staged hydrazine gas generator had demonstrated satisfactory performance with a small thermal control pack designed for 1650°F. Subsequent activities were directed to reduce the output temperature to 1450°F, requiring a considerable enlargement of the thermal control pack.

The thermal control pack evaluation program was conducted in the existing test position (Cell 106, Rockwell Thermodynamics Laboratory) which was used for all of the fast start tests.

All tests were targeted for the design conditions (710 psia chamber pressure, 1450°F chamber temperature, and an 8.34 lb/sec hydrazine flowrate), and a 10-second duration. The firing sequence was automatically controlled and monitored for safe operation. The manual activation of the pilot propellant supply valve initiated the remainder of the firing events. A temperature controller issues the "open" command for the first stage valve when the pilot output temperature reaches the 1200°F value. The opening command for the second stage valve is generated by the sequencer 300 milliseconds after the first stage valve is signaled. Both the first and the second stage chamber pressures were monitored to automatically terminate the firing if the initial start transient for each stage failed to result in a normal chamber pressure buildup rate.

The gas generator hardware used in this program was, for the most part, existing hardware, as used in the preceding programs. The exceptions are in the cases of two catalyst packs involved. A new pilot pack was fabricated because of the damage incurred to the perforated "floating " plate at the exit end of the pilot pack. The new pilot pack was quite similar, with some design modifications introduced to reduce catalyst loss rates and improve the retention system for the perforated end plate.

A new thermal control pack was also fabricated for this program. The existing design was changed to improve its structural integrity. Additionally, finer mesh size Shell X-4 catalyst (20/30 mesh) was used instead of the 8-12 mesh catalyst previously employed. The increased particle surface area and more constricted gas flow paths resulted in a higher degree of decomposition of the NH_3 component of the exhaust gases. This endothermic reaction was required in order to lower the exhaust temperatures to the 1450°F target value, or lower.

The initial catalyst pack is shown in Figures 21 and 22. The inner and outer walls of this conically shaped pack are fabricated from INCO 600 wire screen material. The outer wall consisted of an initial layer of 60 mesh X 0.008 inch wire diameter screen, whose function is to retain the catalyst fines. This layer was followed with a 40 mesh X 0.010 inch diameter screen, also used for particle retention. The final (outside) layer was formed from 5 mesh X 0.080 inch diameter wire screen. This layer constituted the structural wall. The inner wall of the pack is of similar construction except that it consists of only a 40 mesh screen and a 5 mesh screen.

At the beginning of the current test phase it was determined by data review and extrapolation that the thermal control pack should be enlarged to contain approximately 6.4 lb. of Shell X-4 catalyst, and that the catalyst particle size should be made finer, 20-30 mesh as compared to 8-12 mesh previously used. The large catalyst carrier, and the fine mesh catalyst,

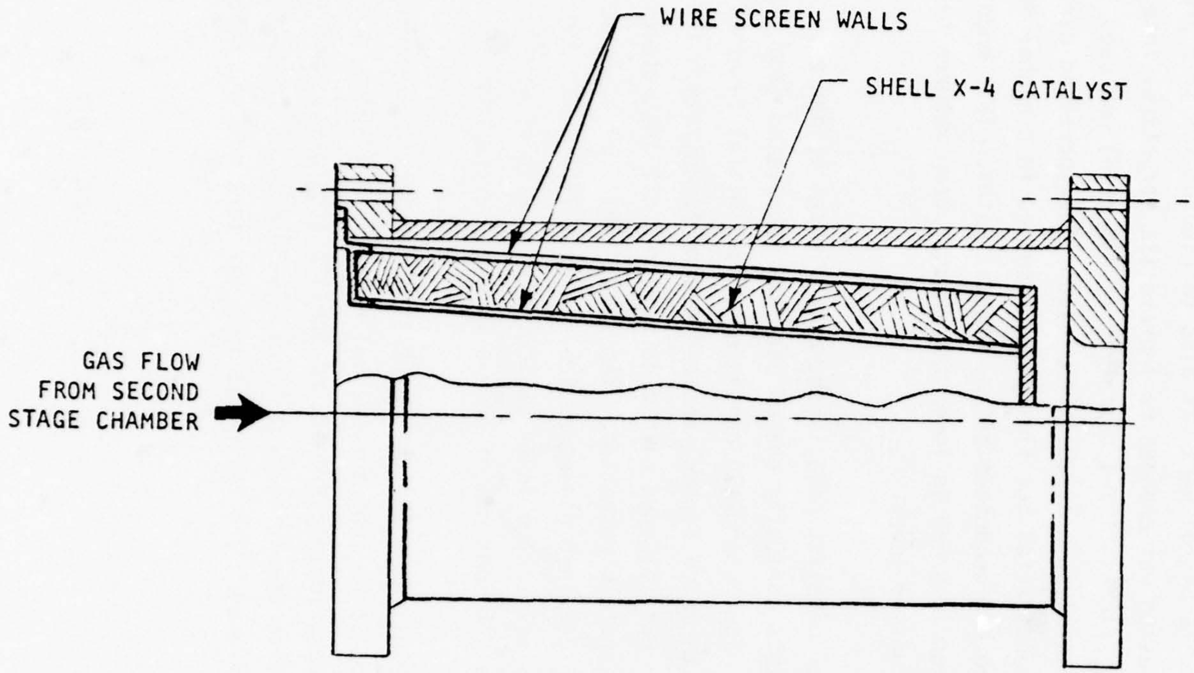


Figure 21. Plenum Chamber with Thermal Control Pack

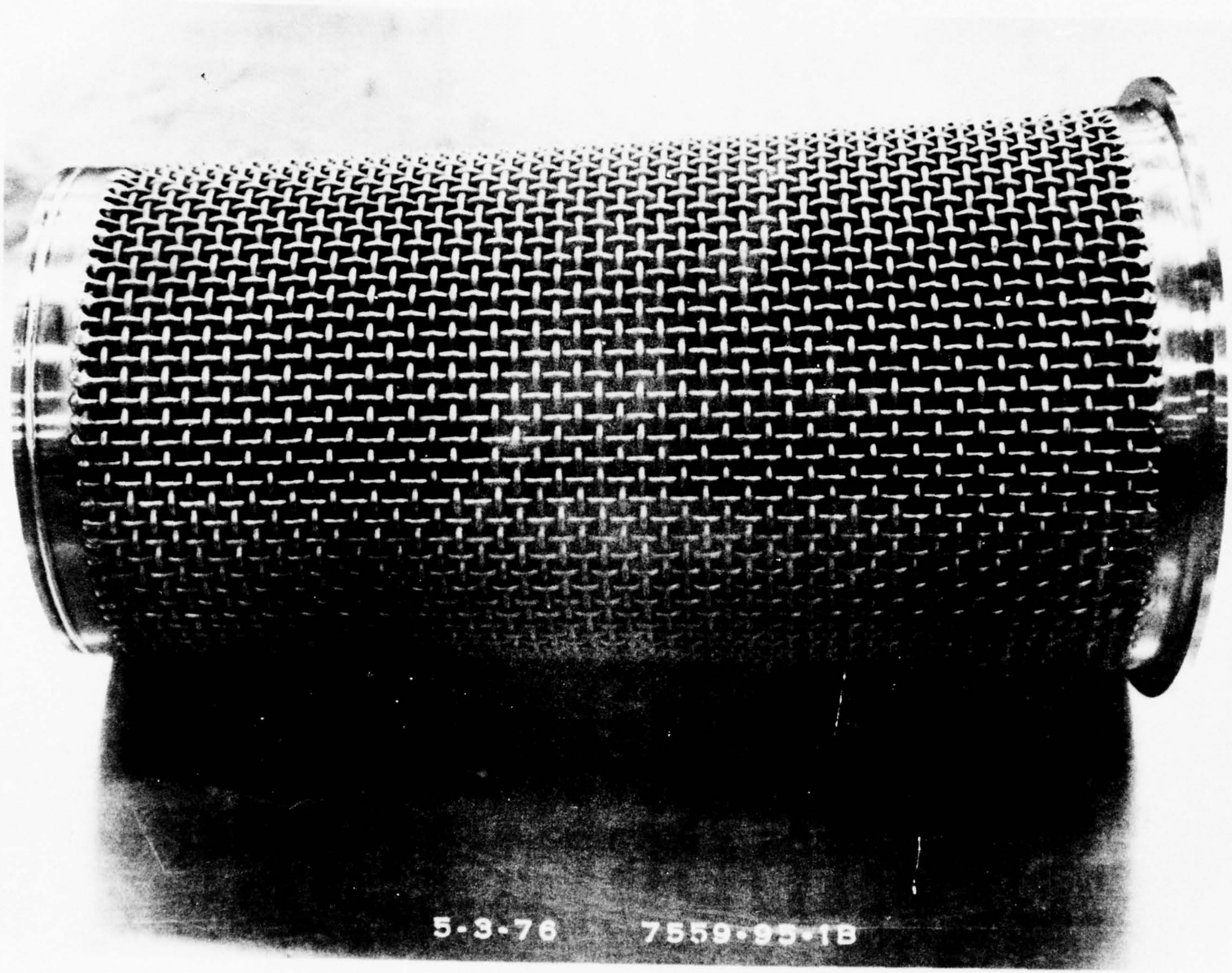


Figure 22. Thermal Control Pack Initial Design (Pack IV-1) Not Filled with Catalyst and Without Cover Plate

would have an unknown effect on pack durability, but it was suspected that the life would be affected unfavorably. In some of the previous programs, the copper gaskets used with the breadboard hardware has exhibited incipient melting, so it seemed prudent to experiment with higher-melting material.

In the course of the test program, 41 tests were made, all showing good repeatable operation of the staged gas generator. Output gas temperatures were reliably obtained averaging 1340°F, a result favorable to eventual interpolation to 1450°F. The result is also favorable if a wider temperature range is acceptable, since the output temperature would be expected to gradually increase as the catalyst deteriorated. A wide range will give the longest useful life.

In the course of the 41 test program, failure of the structural parts of the catalyst carrier was experienced three times. Each time the hardware was rebuilt with minor improvements. A fourth iteration in the carrier design was implemented as turbine testing proceeded (subsequent to this task) resulting in a nominal carrier screen life of about 30 starts between overhauls.

The existing 17 element (0.021 inch diameter injection orifices) injector, with a thermal standoff between the manifold and body, was used for this program.

A number of nickel gaskets were fabricated and installed for evaluation. These gaskets were candidates for replacement of the copper gaskets previously used as the hot gas seals at the various flange joints .

Copper is mechanically quite suitable for this purpose, but has some compatibility problems with hydrazine and its decomposition products. Some evidence has been noted of traces of copper migrating through the gas generator system and being deposited upon the walls of the turbine first stage nozzles. Nickel, in addition to being chemically much more inert to the gas generator fluids, has a considerably higher melting point than copper,

2647°F vs 1982°F, so that considerably more margin would also result. The evaluation demonstrated that the nickel gaskets are not soft enough so that the machined serrations in the steel flange joints will be impressed into the gasket sufficiently to result in an effective hot gas seal.

E2 Performance Tests [12, 13]

The major objective of this program task was to demonstrate that the forthcoming E3 turbine could deliver the specified performance. The E2 task was initiated by contract MOD P00008, which provided new criteria for the design and by MOD P00012, which authorized fabrication of subsequent turbines using more conservative subsonic rotor aerodynamics for the first stage nozzles and rotor blading.

Eleven tests were performed during June 1976, using heated air rather than use the gas generator because:

(1) The integrally bladed E2 wheels were not considered dynamically safe at high speeds due to high speed blade resonance identified by holographic test at AFAPL in 1976.

(2) Lower speed air driven operation properly models turbine characteristics such that good stabilized (longer test duration) data points can be obtained to define performance.

Secondary objectives included evaluation of revisions to the hydrostatic thrust bearing, and evaluation of water brake modifications expected to improve facility calibration accuracy.

The test facility supplies heated air at 800°F (1260°R), 220 psia and 4.95 pounds per second to the turbine inlet flange. Since the turbine nozzle is choked, the supply conditions remain constant at any turbine speed. The turbine outlet pressure for the supply and turbine design pressure ratio is 5.34 psia which the facility can achieve. The turbine isentropic spouting velocity for air for the inlet total pressure and temperature and outlet static pressure was determined and multiplied by the design velocity ratio to determine the design equivalent speed of 15,000 rpm.

Test Installation - For the subject performance evaluation series the hydrazine gas generator was removed and an 8-inch insulated pipe was installed to the turbine inlet from the facility air supply. The facility air supply is equipped with a natural gas-fired heater capable of producing air at the required flow and 800°F. A 4-inch control valve was installed in the 8-inch air line. The facility schematic is shown by Fig. 22. The main exhaust is ducted to the facility vacuum installation and an emergency explosive-actuated bypass valve connects the turbine inlet with the exhaust as a second protection against turbine overspeed.

Prior to test, the 26-foot diameter sphere shown, together with all of the associated ducting systems, is evacuated to an approximate 80,000 feet altitude pressure condition. During test, the total of approximately 25,700 cubic feet of evacuated volume is available to absorb the turbine exhaust air. The Kinney vacuum pump system also operates, so that some of the turbine exhaust air is continuously removed. However, because of the limited vacuum pump capacity (approximately 4,000 cu ft/min), the system operates essentially in a "blowdown" mode. Sonic orifices are used in the turbine exhaust duct for the 25 pressure ratio tests to maintain a fixed turbine exhaust collector pressure until the facility exhaust system pressure builds up to the point where sonic flow no longer can be maintained.

Because of the low powers to be measured by the 6000 hp dynamometer, a smaller (2-inch) control valve was installed in the water circuit. The dynamometer load cell installation was also improved by the addition of flexures to eliminate side load effects.

Test Log - A series of 11 tests was made, starting on 3 June 1976, and ending on 18 June. During the first 5 tests, difficulty was experienced in obtaining the required air temperature, even with prolonged preheating of the system. Accordingly, the facility air system was modified by installing additional heat insulation and by installing a bleed valve at the main air control valve to maintain a flow of heated air during the last minute run preparations. After these modifications, the air system worked well, and in fact, temperatures up to 1013°F were attained.

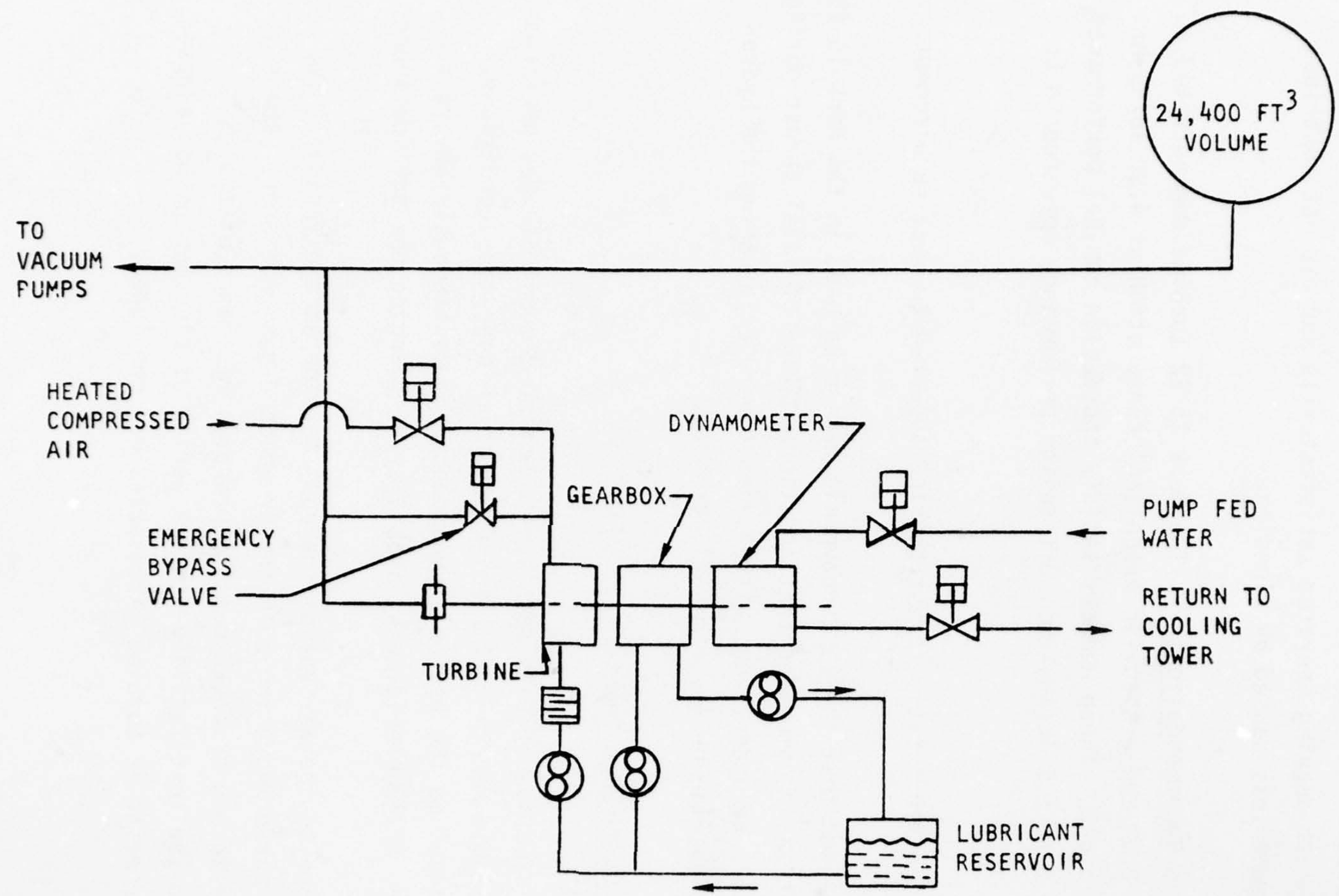


Figure 23. E2 Test Schematic

Hardware - A secondary objective of the performance test series was to checkout some modifications to the hydrostatic thrust bearing to improve performance, operation and life. The hardware changes listed on Fig. 24 were incorporated in the Mark 15-E2 built for an operating checkout. This refurbished thrust bearing operated satisfactorily and did not contribute measureable power loss caused by flooding.

Conclusions and Recommendations - The Mark 15-E2 turbine demonstrated an efficiency of 0.68 and a specific propellant consumption of 4.90 lb/hp-hr at rated conditions. These numbers satisfy the design nominal performance criteria of 5.0 to 5.5 lbs/hp-hr. Off-design performance appeared to be predictable.

The E2/E3 design hydrostatic thrust bearing appeared to meet requirements.

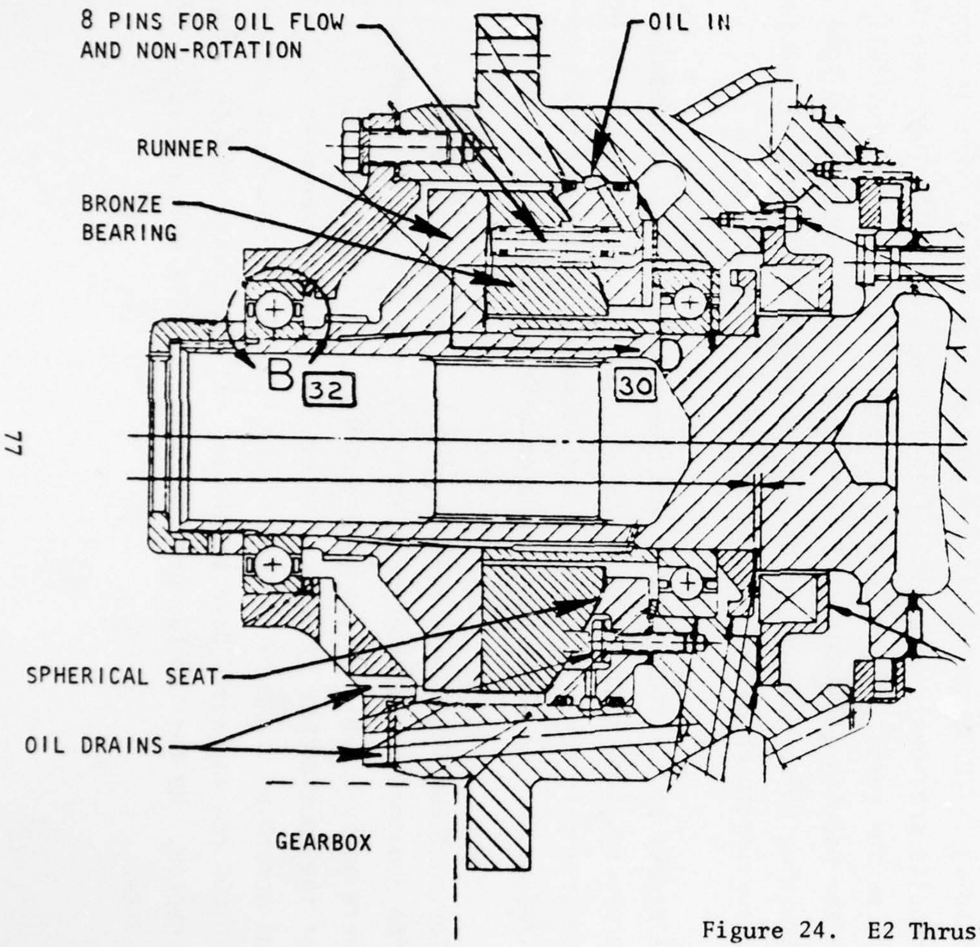
It was recommended that the E2 aerodynamic design be used in the Mark 15-E3 turbine and it was recommended that additional study of axial thrust during start transients be undertaken, with a view toward eliminating the hydrostatic thrust bearing or the rim seal, or both.

E3-1 Turbine Acceptance Tests [16, 17]

The primary objective of this test series was to demonstrate the performance of the deliverable Mark15-E3-1 turbine at design operating conditions. These tests required the use of a dynamometer to continuously absorb the turbine power, while providing a capability of changing the turbine speed.

Tests - Prior to the arrival of the E3-1 turbine at the facility, two firings were performed using only the breadboard gas generator. These tests were performed to check out the proper performance of the gas generator and the test facility since a period of five months had elapsed since the preceding E2 turbine test series was concluded.

HYDROSTATIC THRUST BEARING



1. DISH RUNNER 0.001 TO COMPENSATE FOR CENTRIFUGAL DISTORTION.
2. ADD 8 PINS TO CARRY OIL PRESSURE ACROSS THE SPHERICAL SEAT, ONE PIN FOR EACH BEARING PAD. PINS ARE ORIFICED.
3. DELETE ANTI-ROTATION PIN. (FUNCTION SERVED BY 8 OIL PINS).
4. SUPPLY OIL JET FOR INNER TURBINE BEARING.
5. INCREASE AREA OF OIL DRAINS.
6. REMOVE SLOTS FROM BEARING PADS.

Figure 24. E2 Thrust Bearing

An existing section of ducting, which replaces the turbine, was connected between the facility flange for the turbine inlet duct and the turbine exhaust collector housing. An 1.673 inch diameter exit orifice (Mark 15 E-0 turbine nozzle throat area simulation) was an integral part of the piping. This resulted in somewhat lower gas generator chamber pressures at design flowrates than would exist with the smaller throat area of the Mark 15-E3 turbine, but did not affect the basic objectives of the firings.

The chamber pressures noted and the plenum chamber temperature were consistent with previous results, and it was concluded that no significant instrumentation or operational problems existed.

The catalyst trap installed downstream of the temperature control pack did not show any deterioration as a result of these firings and the temperature reduction pack was in good condition following the gas generator checkout firings. An approximate 1.7 percent weight loss of the 6.385 lb. initial loading of 20/30 mesh Shell X-4 catalyst was noted. This is comparable with results previously observed. This initial weight loss has been interpreted to be the result of the loss of catalyst fines that may initially exist after packing and additional fines that may be generated by some catalyst crushing which could occur because of control pack special readjustments occurring in going through the initial temperature cycle.

A series of six stall torque tests were made with the dynamometer shaft prevented from rotating by means of a locking bar. The stress levels occurring in the bolts holding the locking bar to the dynamometer shaft limited turbine flowrates to approximately 1/3 of the design flowrate.

During these stall torque tests, the turbine/gearbox lubrication systems, the dynamometer lubrication system and the dynamometer waterflow systems were active. This fail-safe procedure was followed so that if a failure of the locking bar system was encountered, the dynamometer would be capable of absorbing the turbine power. The turbine overspeed cut was set at 10,000 rpm and would have terminated all operations when that speed was reached. No locking bar problems were actually encountered during these tests.

Runn 2.2 was made with only the pilot and Stage 1 gas generator sections operating and was primarily a checkout run. All five of the later runs scheduled all-stage gas generator operations.

In order to limit the gas generator second stage flowrate to the desired level, the cavitating venturi upstream of the Stage 2 main valve was throttled. Reduced propellant flowrate resulted in a much longer prime time for the Stage 2 propellant feed system downstream of the cavitating venturi. Runs 2.2 through 3.3 were terminated shortly after start of Stage 2 chamber operation by the Pc-2 pressure level comparator because of the slow Stage 2 chamber pressure buildup. The beginning time for the comparator sampling was successively delayed until the setting was found which would ascertain that proper ignition had been obtained, while at the same time, allow for the long system priming transients. Run 3.4 achieved the scheduled duration of approximately 10 seconds of all stage gas generator operations and the desired stall torque data under stabilized test conditions were obtained during this firing.

Five turbine performance test firings and 6 turbine gas spin tests were made during this series. Because of the highly important need to know the turbine speed accurately, both for performance calculations and for the overspeed cut safety functions, the following actions were taken:

(1) Replacement of the installed magnetic pickup

(2) Installation of a photoelectric pickup device to independently measure the dynamometer shaft speed

(3) Addition of another comparator to monitor the photoelectric speed pickup signal so that a completely redundant overspeed cut system would exist

(4) Installation of a high flow GN_2 system so that the turbine could be driven at a preselected low speed to checkout proper operations of all systems before committing the turbine to a hot firing.

The series of 6 turbine spins were performed to calibrate the GN_2 turbine drive system and to verify proper functioning of both speed monitoring systems.

Hot fire tests resulted in data at conditions ranging from 85 to 103 percent of the design speed level and 91 to 104 percent of the design power level. Test 6.4 was an approximate 2-second scheduled duration, full turbine power hot firing. Data from this initial hot firing was analyzed for proper operation of the speed measuring systems before proceeding to longer duration firings.

The duration of the all stage portion of the tests was reduced from the 10 seconds used previously because of a noticeable bulging of the uncooled gas generator Stage 2 chamber walls. Since this growth has occurred over a series of 61 firings of 10 second duration, a high probability of failure of the chamber did not appear likely, but it was deemed prudent to reduce the run duration to a minimum useable range.

Test 7.3 was a 6.6-second duration firing at approximate turbine full power conditions at a turbine inlet temperature of 1285°F compared to the design 1450°F. The temperature control pack was removed and repacked with 8/12 mesh catalyst instead of the 20/30 mesh size.

Test 8.3 was subsequently performed at a slightly higher gas generator flow-rate and with an increased turbine inlet temperature (1485°F). Since the analyses of the data indicated that the program objectives had been met, the turbine was removed for a teardown inspection prior to delivery. During this inspection, it was found that the inboard ball bearing supporting the turbine shaft had failed during the tests. The problem occurred because of an erroneously tight fit between the outer race of the bearing and its mating housing bore. This race must be free to move axially in response to changes in the thrust bearing gap dimensions with changes in the turbine thrust loading.

The turbine was subsequently reassembled and test 10.2 was conducted at full turbine power with a 7.4 second duration. This test also demonstrated that the turbine developed the design 6,000 horsepower, and the turbine was removed for a final inspection and delivery for use by AiResearch in their experimental program. All of the test hardware and facility items were found to be in good condition at the end of the tests.

Results - The performance characteristics of the Mark 14-E-3-1 turbine were calculated primarily from run numbers 8.3 and 10.2 test data. By taking a number of data slices for each of these runs, turbine performance was characterized over a range of 85 to 103 percent of the 29,000 rpm turbine design speed and a range of the design 6,000 horsepower power level.

The turbine performance parameters presented in Table 9 are all based upon the use of the developed horsepower as indicated by the dynamometer load cell data. It should be noted that corresponding horsepower data is also listed which is based on the dynamometer water throughout and its temperature rise (which was measured with a differential thermopile setup). This thermal method of calculating the developed horsepower tends to indicate somewhat higher values, and is usually within two percent of agreement. These considerations tend to confirm the validity of the dynamometer output force calibration technique and indicate that the turbine horsepower used in the calculations may be somewhat conservative.

One other additional factor is known which tends to make some of the performance results lower than the true values. This is the face that the data slices which were used for performance calculations at turbine speeds of 29,000 rpm and higher, are not steady-state data slices. At these times, the turbine was being accelerated and torque was being absorbed by the turbine, gearbox, and dynamometer rotating members. For example, during run 8.3, the turbine was accelerating at an average rate of 87 radians/sec^2, and during run 10.2, the acceleration rate was approximately 419 radians/sec^2.

TABLE 9 E3-1 TURBINE PERFORMANCE

Parameter	Run 3.4	Run 7.3 Slice 1	Run 7.3 Slice 2	Run 7.3 Slice 3	Run 8.3 Slice 1	Run 8.3 Slice 2	Run 8.3 Slice 3	Run 10.2 Slice 1	Run 10.2 Slice 2	Run 10.2 Slice 3	Run 10.2 Slice 4	Run 10.2 Slice 5
Turbine Speed, RPM	0	28,960	29,465	29,840	27,166	29,626	29,016	25,368	25,547	24,647	29,039	29,729
Inlet Temp. (Total) - F	1030	1272	1278	1285	1452	1484	1479	1457	1468	1474	1482	1483
Flowrate - lb/sec	2.068	8.248	8.218	8.284	8.422	8.431	8.445	8.598	8.604	8.600	8.604	8.619
Inlet Press. (Total) - psia	163.6	668.3	670.2	672.4	693.4	702.3	701.6	713.1	716.9	718.9	722.0	722.8
Exhaust Press. (Static) - psia	8.347	16.60	16.48	16.83	16.19	17.06	16.97	17.55	17.68	17.80	17.41	17.31
Δh_a T-S - Btu/lb	561.4	716.5	719.1	719.0	768.9	772.1	771.3	763.9	766.3	767.3	772.4	773.5
Pressure Ratio P_{T1}/P_{S2}	18.40	40.28	40.67	39.95	42.83	41.17	41.34	40.63	40.55	40.39	41.47	41.76
Um/CO T-S	0	.2594	.2635	.2669	.2349	.2557	.2505	.2201	.2213	.2134	.2506	.2563
HP Dynam. (Torque)	0	5,288	5,431	5,468	6,074	5,939	5,854	5,900	5,993	5,966	5,915	6,064
HP Dynam. (Water ΔT)						5,997	5,939	5,995	6,119	6,041	6,243	6,110
HP Gearbox	0	-143	-147	-148	-155	-157	-154	-148	-150	-147	-155	-160
HP to Gearbox	0	5,431	5,578	5,616	6,229	6,096	6,008	6,048	6,143	6,113	6,070	6,224
SPC to Gearbox, lb/hp-hr	0	5.467	5.304	5.310	4.867	4.979	5.060	5.118	5.042	5.064	5.103	4.985
HP Ball Bearings	0	-13	-13	-13	-12	-13	-13	-10	-10	-10	-13	-13
HP Thrust Bearing	0	-230	-244	-256	-209	-273	-255	-172	-175	-160	-249	-267
HP Seal	0	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
HP Disk Friction	0	-9	-9	-10	-7	-9	-9	-6	-6	-5	-9	-10
HP Blading	0	5,684	5,845	5,896	6,458	6,392	6,286	6,237	6,335	6,289	6,342	6,515
SPC Blading, lb/hp-hr	0	5.224	5.062	5.058	4.695	4.748	4.836	4.963	4.889	4.923	4.884	4.763
Torque - Ft-Lb	398.3	958.4	968.1	962.6	1174.3	1052.9	1059.6	1221.6	1232.1	1271.3	1070.0	1076.4
Axial Thrust - lb	-201	254	256	188	-110	-127	-156	527	530	471	701	754
Calculated A_c - in^2	1.632	1.568	1.559	1.568	1.585	1.574	1.577	1.575	1.570	1.566	1.562	1.563
Efficiencies												
η Blading - Test	0	.6798	.6990	.6996	.7048	.6940	.6821	.6711	.6791	.6736	.6745	.6907
η Blading - Predicted	0	.6739	.6775	.6803	.6478	.6704	.6654	.6282	.6299	.6183	.6654	.6711
η Test/η predicted	0	1.009	1.032	1.028	1.088	1.030	1.025	1.068	1.078	1.089	1.014	1.029
η/U/CO Test	4.562	2.621	2.653	2.621	3.000	2.714	2.723	3.049	3.069	3.156	2.691	2.695
η/U/CO Predicted	4.200	2.598	2.571	2.549	2.759	2.622	2.656	2.854	2.846	2.897	2.655	2.618
η/U/CO Test/Predicted	1.086	1.009	1.032	1.028	1.088	1.030	1.025	1.068	1.078	1.089	1.014	1.029

The limitations in the allowable run duration (because of the deteriorated condition of the breadboard gas generator), and the necessity to spend a minimum time in the high speed range because of possible Stage 1 blading torsional vibration problems, dictated this method of conducting the tests. All of the data points at 29,000 or higher speeds could be corrected for inertial torque losses, but this has not been done.

	DESIGN	TEST	
		Run 8.3	Run 10.2
Inlet temperature, °F	1450	1479	1482
Inlet Pressure, psia	710	702	722
Flowrate, lb/sec	8.33	8.44	8.60
Efficiency	0.68	0.68	0.67
Turbine shaft power, HP	6000	6008	6070
Speed, rpm	29,000	29,016	29,039
Specific propellant consumption lb/hp-hr (nominal/max.)	5.0/5.5	5.1	5.1

The test data indicate that the Mark 15 E3-1 turbine met or exceeded the design specifications in all respects. The results presented are very conservative in that the lower of the two turbine horsepower determinations is presented (load cell indicated horsepower and dynamometer water ΔT indicated horsepower) and the data was not corrected for inertial torque losses.

The Mark 15 E3-1 turbine was found to be in good condition after the tests were completed indicating that the bearing problem initially encountered had been properly resolved.

E3-2 Turbine Performance Tests [18, 19]

The primary objective of this test series was to demonstrate that the E3-2 turbine had been beneficially modified (relative to the E3-1) to increase its allowable upper speed limit from 103% to 110%. Further turbine performance characterization and demonstration were also required.

Tests - Two gas generator tests were conducted on 8 November 1977, establishing initially that first stage ignition was proper. The second generator test with all stages operational served to establish that the system which had been out of use since for five months was ready for turbine test. A total of 10 cold gas spin tests were also made during this E3-2 series to provide pre and post hot fire checkout of instrumentation, facility systems and dynamometer calibrations.

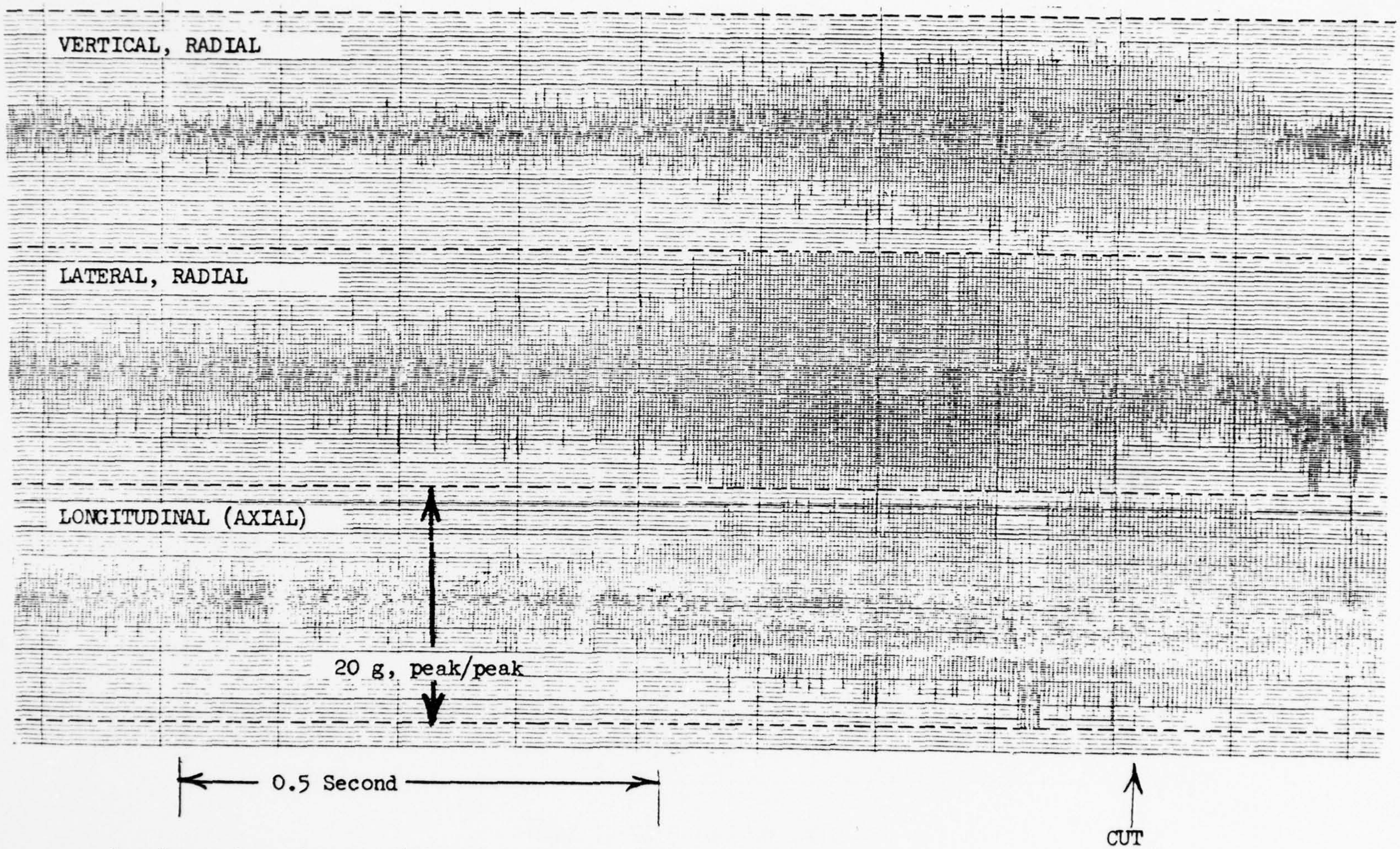
Three hot fire tests (#12.5, #12.6 and #14.1) were made through 2 December 1977, followed by a second set of six hot fire turbine tests in between 26 and 30 January 1978. Significant useable performance data was obtained from all nine hot fire tests. The year-end gap in testing between 2 December 1977 and 26 January 1978 was required for corrective action to eliminate a self-excited "whirl" mode of instability which was exhibited during test 14.1 on 2 December 1977. Tests during January served to demonstrate that the instability was resolved. Turbine E3-2 was disassembled after the final test of the series, found to be in excellent condition, and then rebuilt and stored pending allocation for other objectives.

Results - Table 10 displays the accumulated performance data of the E3-2 test series and Fig. 25 and 26 display the E3-2 accelerometer readings during test 14.1 instability and the E3-2 stable accelerometer readings during test 18.4 under nearly identical test conditions. Although several whirl problem corrective actions were taken between the two sub-series of tests to ensure that all possible contributors had been foiled all indicators pointed to a single most probable cause for the problem. Of the five major types of whirl only hysteretic whirl seemed likely and is attributable to looseness in the turbine rotor stack. Following test 14.1, an unacceptably loose condition was found and corrected by design modification and assembly methods. All of the performance data for the E3-2 turbine satisfied the design criteria and correlated well with E3-1 data.

TABLE 10

MARK 15-E3-2 TURBINE DATA

	TEST NUMBER						
	14.1	14.2	18.1	18.2	18.3	18.4	
SPEED	30155	30226	26121	31502	27943	31818	RPM
	104.0	104.2	90.1	108.6	96.4	109.7	%
POWER (DYNAM)	6391	6414	5768	6082	4942	5921	SHP
	106.5	106.9	96.1	101.4	82.4	98.7	%
SPC	4.72	4.67	4.95	4.77	5.06	4.87	#/HPH
INLET TEMP	1544	1547	1514	1516	1484	1513	°F
INLET PR	722	723	717	728	625	729	PSIA
EXIT PR	16.8	16.8	18.3	17.4	16.7	17.4	PSIA
PR. RATIO	43.1	43.1	39.2	41.9	37.4	42.0	-
$\triangle h$ T-S	792	792	773	782	760	781	BTU/#
	DECEMBER		JANUARY				



Accelerometer signals filtered to remove frequencies above 600 Hz

Figure 25. Conclusion of Run 14.1, Accelerometer Traces Showing Burst of Sub-synchronous Whirl

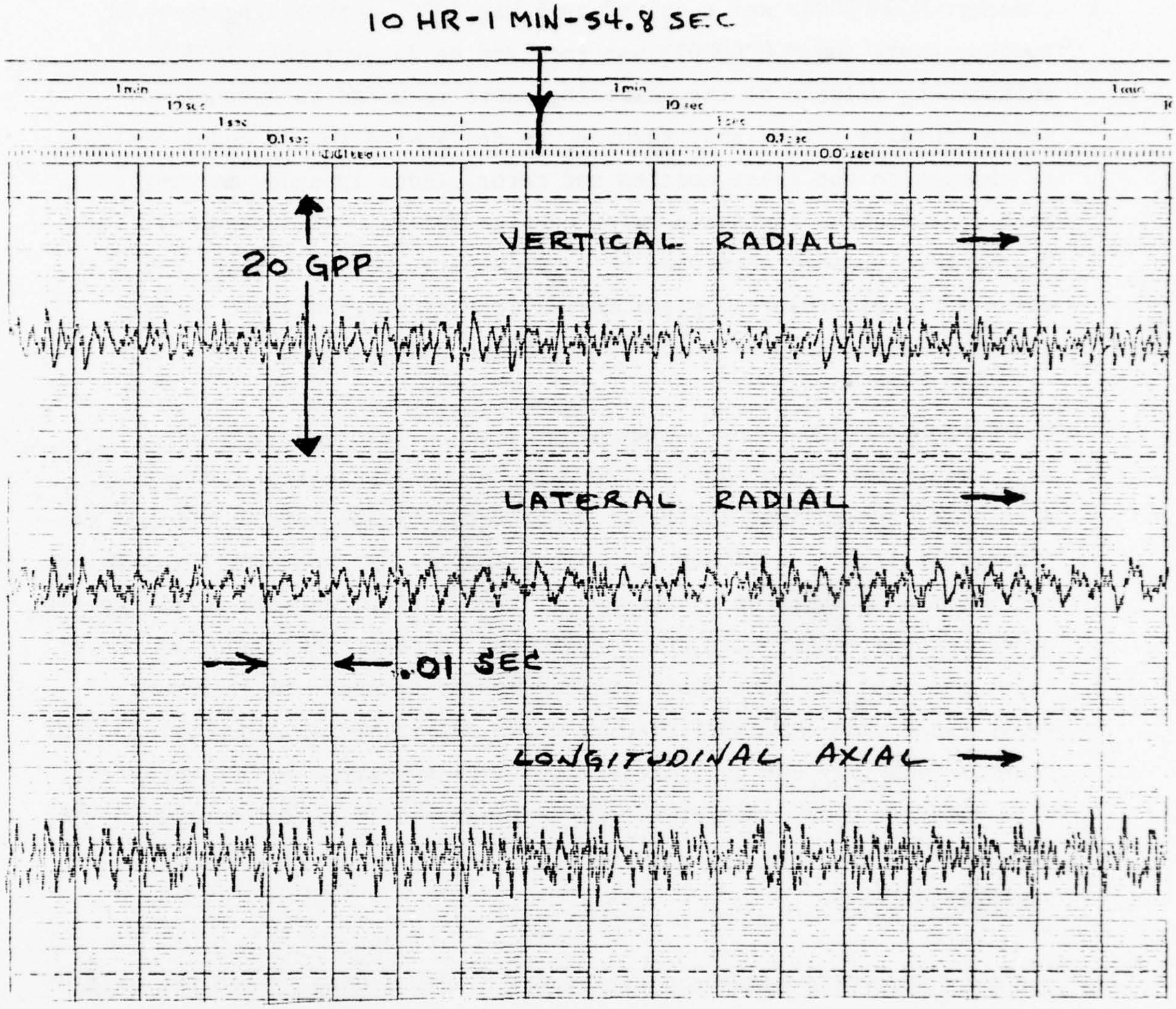


FIGURE 26. MARK 15-E3-2 TURBINE ACCELEROMETER DATA TEST 18.4 AT 105% SPEED

*NOTE: THIS 0.18 SEC. DATA SLICE IS TYPICAL FOR THE SUBJECT TEST OF 9.0 SEC.

4. DESIGN ITERATIONS (E2/E3) [14, 15, 19]

Although only one test turbine had been planned in the basic program, contract MOD P00008 was received on 5 May 1975, authorizing some E2 turbine parts and MOD P00012 was received on 25 September 1975, authorizing release of the remaining parts required to assemble a second turbine. This test turbine was designated the E2 model because of changes to the inlet nozzles and rotor blades intended to provide a more conservative (subsonic rotor) design margin. All components for the E2 turbine were completed by late January, but as a result of concurrent E1 test results, plans for E2 testing were deferred pending study of first stage E1 rotor blade damage problems.

The E1 rotor blades were subjected to shaker tests at AFAPL using holographic instrumentation and at Rocketdyne using conventional instrumentation. Both tests agreed that the first stage rotor blades were moderately responsive at 11,000 to 14,000 Hz. Since four recent tests of the turbine had included a substantial amount of steady state run time in the corresponding speed range of 16,000-20,000 RPM (a value influenced by the 41 inlet nozzles) adequate cause for damage was offered. A 12,354 Hz first blade mode was identified by these shaker tests with about 1000 Hz scatter (possibly due to symmetric and anti-symmetric bending modes). The symmetric modes consist of combinations of (1) blade bending in the direction of the axis of symmetry and (2) bending of the free corners symmetrically.

The anti-symmetric modes consist of (1) blade bending 90° to the axis of symmetry (2) blade torsion and (3) bending of the free corners anti-symmetrically.

There was no evidence of significant E1 resonance modes at higher frequencies indicating that the MK15-E1 turbine was suitable, as designed and built, for its intended operational range of 26,100 to 31,900 RPM.

The new subsonic E2 integrally bladed turbine rotor was subjected to similar tests which established that the first stage blades were strongly resonant at about 19,469 Hz - corresponding to 28,491 RPM with the 41 nozzle installation. As a corrective action, work was initiated to define an E3 rotor wherein the solution to the problem consisted of fabricating individual, shrouded blades to the E2 aerodynamic design, which were then fixed to the rotor discs using fir-tree-root processes.

Figure 27 describes the vibrational modes of the new (E3) rotor blade configuration and Fig. 28 describes the E3 test results at Rocketdyne and AFAPL during 1976 with modes related back to Fig. 29. The critical 19,469 Hz torsional mode was driven upward to 22,282 Hz by the modifications (corresponding to 32,607 RPM - with 41 inlet nozzles). Figure 29 and 30 display the holographic patterns developed by AFAPL for identification of significant vibrational modes. The new E3 design was released to fabrication in June 1976, leading to assembly of two E3 turbines in January, 1977. The E3 turbine wheel configuration is shown in Figure 31.

During early 1977, it was demonstrated that the E3-1 satisfied all criteria of the specification and was free of significant resonance at its rated speed of 29,000 RPM. The upper speed limit of E3-1 operation was constrained, however, to 29,840 RPM because of concern that there was still not enough safe margin between desired upper speed limit of 31,900 RPM and the stage 1 first torsional mode speed of 32,607 RPM. For the E3-2 turbine, the 41 nozzle assembly was replaced with a new 37 nozzle assembly to push this first torsional mode resonance higher (to 36,133 RPM). After this adjustment was completed the E3-2 turbine was successfully tested to 110% of rated speed (31,900 RPM).

During the cutoff transient of one test at 104% of rated speed a whirl mode of rotor instability was noted and, as a safety precaution, action was taken to eliminate sources of excitation. The most significant of these actions was replacement of a compressible washer in the rotor stack with a rigid washer so that stack bolt torque could not relax during test. The adequacy of these final corrective actions was proven in subsequent tests under identical and limit conditions as discussed earlier in this test narrative.

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HIGH POWER FAST START TURBINE POWER UNIT.(U)
MAR 78 D R HODSON F33615-74-C-2013

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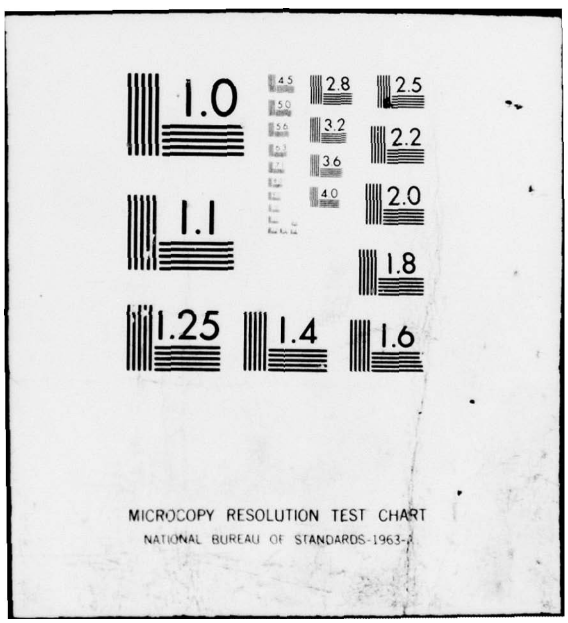
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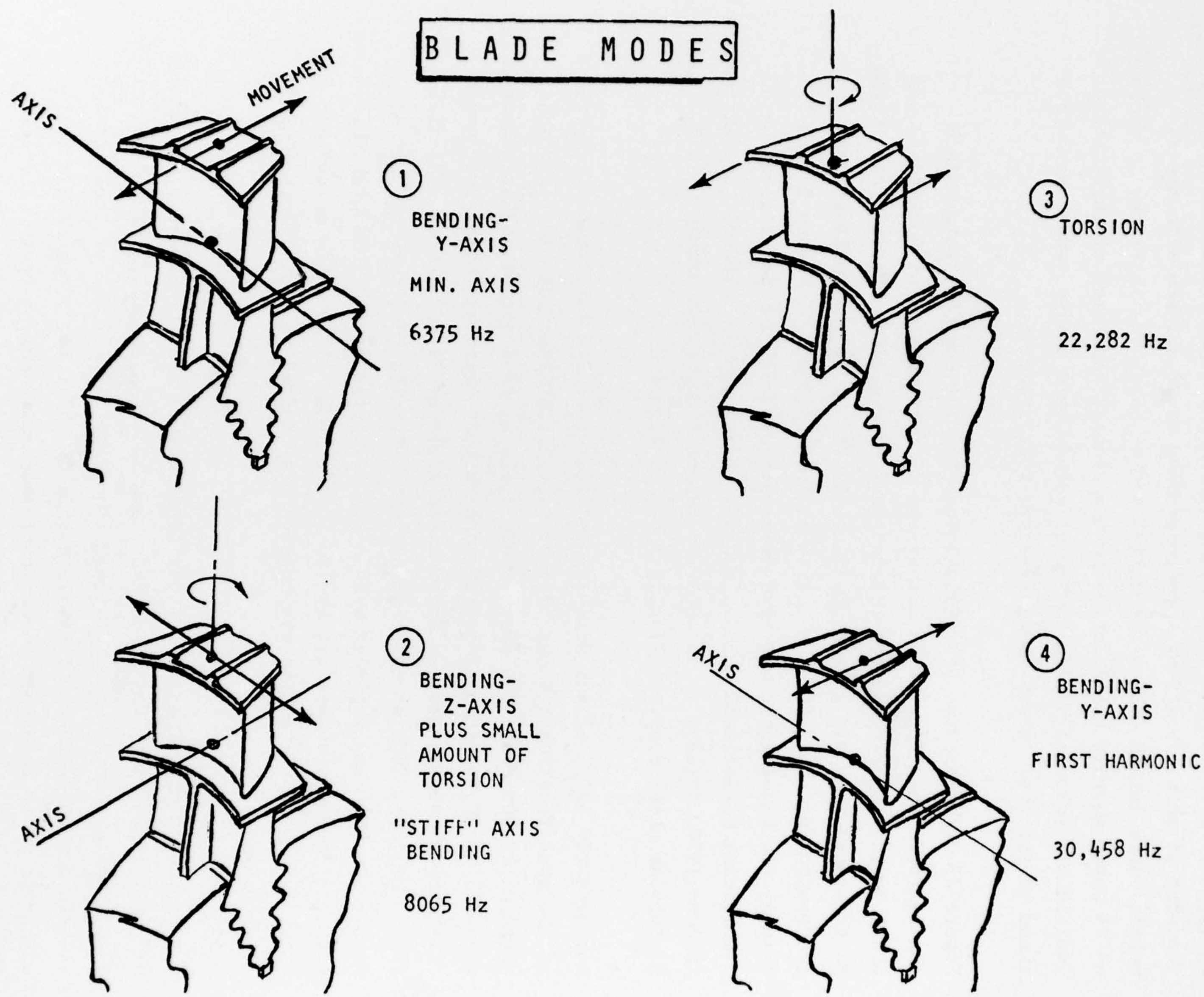


Figure 27. Vibration Modes of the First Stage Blades

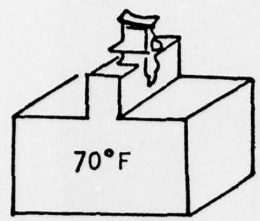
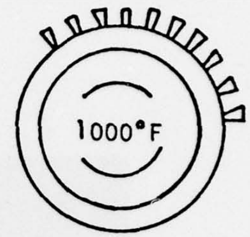
BLADE FREQUENCY	 70°F TEST BLOCK				 1000°F WHEEL AT TEMPERATURE		
			EXPERIMENT				
	MODE	ANALYSIS	R.	APL	MODE	ANALYSIS	BEST ESTIMATE
E3 BLADE	1	6,015	6,000	6,615	1	6,391	6,375
	2	12,350	12,000	12,257	2	8,300	8,065
	3	20,724	22,400	22,600	3	20,615	$\boxed{22,282}$
	4	34,140	32,200	34,174	4	32,293	30,458

Figure 28. First Stage Blade Vibrational Frequencies (H_z) for the Mark 15E-3-1 Turbine

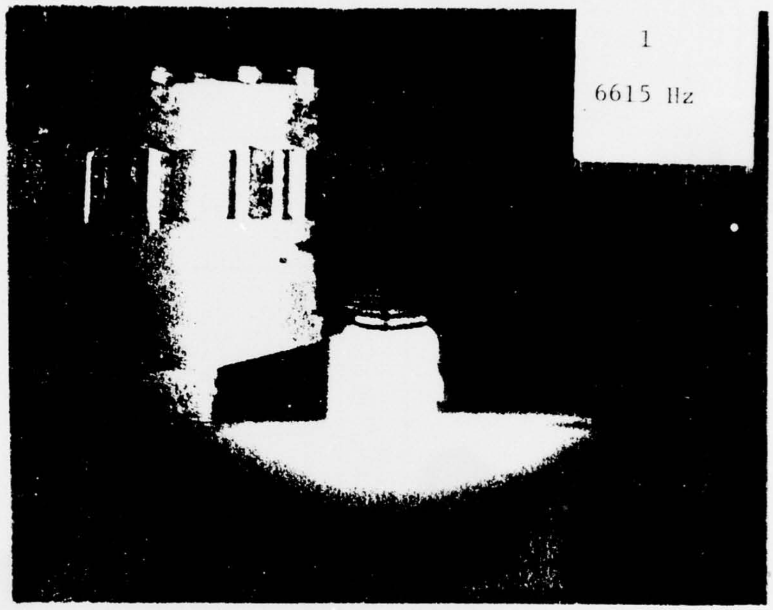


Figure 29. First Flexure Mode - E3 Subsonic Rotor 1st-Stage Blade Resonance

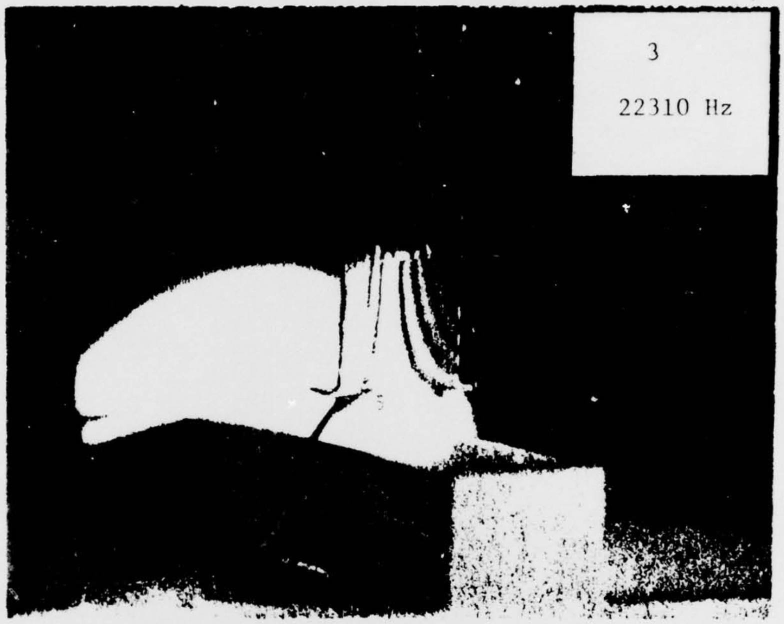


Figure 30. First Torsional Mode - E3 Subsonic Rotor 1st-Stage Blade Resonance

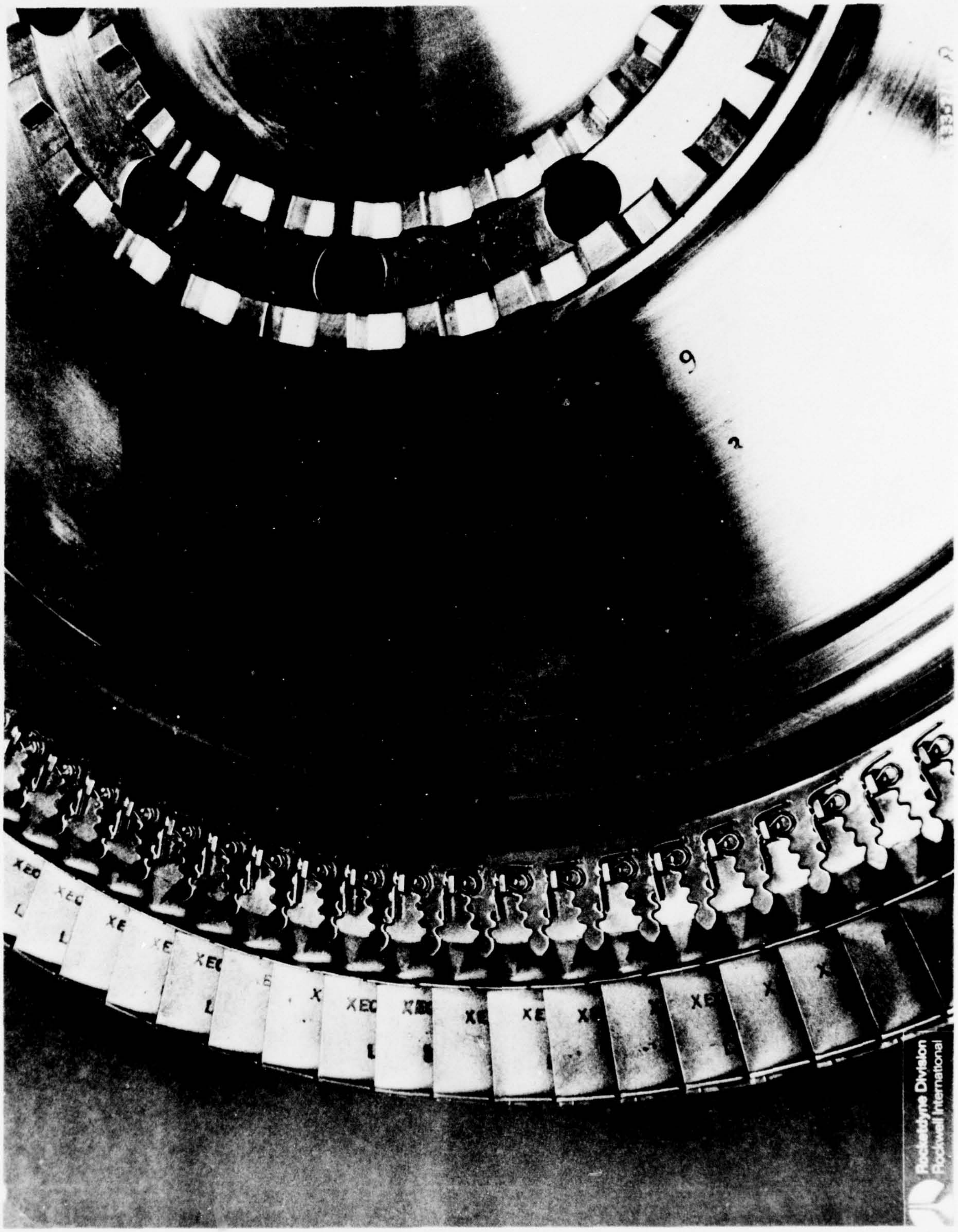


Figure 31. E3-1 Rotor (1st Stage)

SECTION III

TEST EQUIPMENT

The subject test program was conducted in Cell 106 of the Rockwell Thermodynamics Laboratory in El Segundo, California. The major facility revision performed for the present program was the addition of a waterbrake dynamometer high speed gearbox and inertia flywheel discussed below.

1. WATERBRAKE DYNAMOMETER

The waterbrake dynamometer was procured from the Industrial Dynamometer Company of Milford, Connecticut. This unit was designed to be conservatively capable of absorbing 6,000 horsepower at 8,000 rpm. The Mark 15 turbine was coupled to the dynamometer through a 3.2 gear ratio speed reduction gearbox. A photograph of the dynamometer, prior to installation, is shown as Fig. 32.

The dynamometer uses a ball bearing supported central shaft which carries a number of flat, drilled discs. These discs constitute rotors which move relative to stator chambers, which fit closely around the rotor discs. These stator chambers terminate a short distance from the dynamometer centershaft, and are open to this annular space for some distance. In operation, water flows into the rotor chambers at the inner diameter of the stators and is thrown outward by the centrifugal force generated by the rotor action. This water then forms an outer annular ring which is being sheared between the rotor and stator surface. The shear forces results in a reaction torque on the waterbrake housing, which is measured by means of a bonded strain gage load cell. The shear forces dissipate the turbine generated power by conversion into thermal energy, i.e., by increasing the water temperature.

The waterbrake housing is mounted in trunnion bearings and flexible couplings are used in the water pipe connection so that the load cell can sense the full reaction torque. The test installation also monitors the dynamometer water enthalpy change for a second method of power absorption determination.

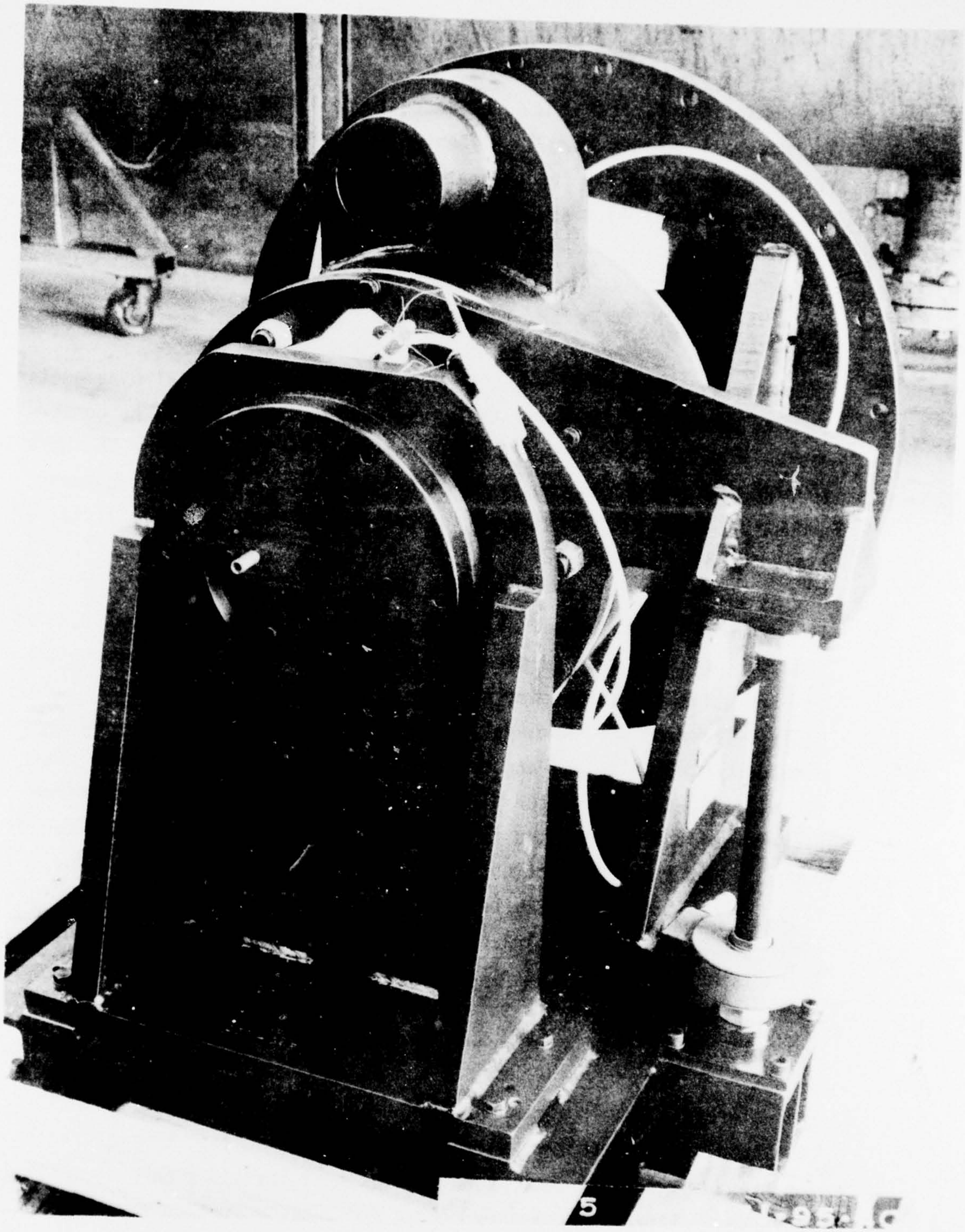


Figure 32. Dynamometer

The water for the dynamometer is pump fed from the facility water cooling tower. After passing through the dynamometer, if flows back to the cooling tower. A nominal 40 psig inlet condition to the dynamometer and 500 to 700 gpm flowrates were maintained. Inlet and outlet conditions can be varied by means of linear-plug throttle valves located in these lines. The valve opening position in each case is commanded by manually adjusting a valve position control knob on the test control console. In the case of the inlet throttling valve, a feedback circuit was also provided so that once the position indicator is set, the valve position will be automatically furtner adjusted to attempt to maintain the inlet pressure in the face of the system transients which occur as the dynamometer goes through its acceleration period.

Steps were taken to make system improvements as follows:

(1) The installation of a torque arm to permit a dead-weight calibration capability for the dynamometer system. (The vendor had provided only a one point calibration of the load cell by itself.)

(2) The installation of a flowmeter in the inlet line of the water supply for the dynamometer. This allowed an alternate computation of the dynamometer power absorption based on the enthalpy change of the water, with its flowrate and temperature rise both being monitored.

(3) The vendor-recommended flexible couplings on the dynamometer water supply lines were replaced with short sections of rubber hose. This was done because there was some doubt as to whether adequate alignment was being maintained between the massive piping and the dynamometer. The vendor-recommended couplings were capable of adjusting to a limited misalignment and, after that point, compression of the internal "O" rings would result in direct contact between metal surfaces.

Following the above described modifications, a multi-point, dead-weight calibration of the complete dynamometer system was performed with the dynamometer pressurized and flowing water at the nominal operating conditions. An "as-is" calibration was also performed, before the system modifications were made, in order to improve the accuracy of the data.

2. GEARBOX

The gearbox employed in these tests had been previously used in the Airborne APU Program. The gearbox was designed to transmit 6,000 horsepower with an input shaft speed of 26,000 rpm and an output shaft speed of 8,126 rpm. The overall gear ratio is approximately 3.2:1. A photograph of the gear train is shown as Fig. 33. All of the gears are spur gears with a 20 degree pressure angle and involute tooth profile. The turbine power is transmitted to a "floating" sun gear and from there to three reverted shafts incorporating pinion gears at both ends. The power is then transmitted out through a second "floating" sun gear, which is internally splined for coupling to the appropriate downstream power absorption component.

The gearbox has a positive lubrication system, with a series of internal jets which are directed at the various gear meshes and at the bearings. The MIL-L-7808 synthetic lubricating oil is supplied to the gearbox at a nominal 90 psi pressure. The gearbox lubricating oil and the oil from the turbine hydrostatic bearing (which is drained into the gearbox) was continuously being withdrawn at the bottom of the gear case by means of a scavenging pump.

During the course of this program, the gearbox was subjected to approximately 100 fast starts and 20.6 minutes of powered operation.

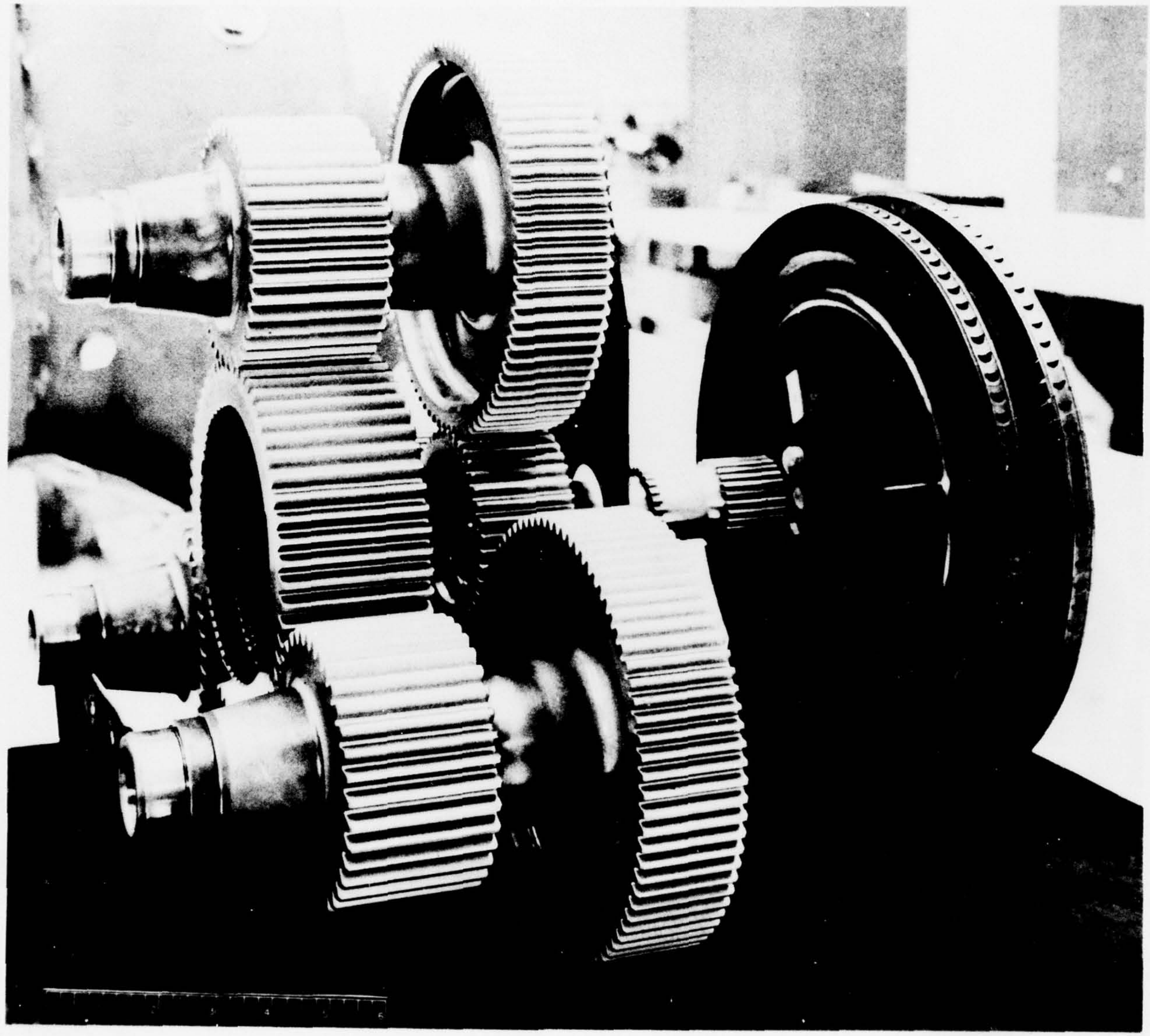


Figure 33 View of Gearing Arrangement for 3.2:1 Gearbox Used Between the Turbine and the Dynamometer

3. INERTIA SIMULATION FLYWHEEL

Inertia simulation for fast start testing was accomplished by the design and fabrication of a special test device for this program as described by Fig. 34. The simulator casing mounts directly to the gearbox and a main rotor is mounted within the simulator on ball bearings with a splined shaft interface to the gearbox. The main rotor disc is 18.0 inches in diameter and 3.2 inches thick resulting in an inertial characteristic of 65 $ft^2\#$ or 2.0 slug ft^2.

Since the flywheel rotates at a speed reduction ratio of 3.2 relative to the turbine shaft its equivalent inertia at turbine (or alternator) speed is reduced by a factor of the speed ratio squared or 10:1. The main flywheel rotor is slotted at its outer edge as shown in the inset of Figure 29 so that facility air or GN_2 can be used to decelerate rotation. A set of attachment discs were also fabricated and the capability was provided to bolt one or both of these discs to the main rotor to increase the total inertia.

Addition of the first extra disc adds 28.3 $ft^2\#$ of inertia and the second extra disc is worth 36 $ft^2\#$ of inertia. Referred to the turbine rotor design speed of 29,000 RPM these discs may be used as follows:

	BASIC FLYWHEEL INERTIA	EFFECTIVE FLYWHEEL INERTIA	TOTAL SYSTEM INERTIA
MAIN FW ROTOR	2.00	.195	.215
ADD 1ST DISC	2.88	.281	.482
ADD BOTH DISCS	4.00	.391	.592

(Slug - $Feet^2$)

Total system inertia includes the turbine rotor, gearbox effects as well as the flywheel rotor.

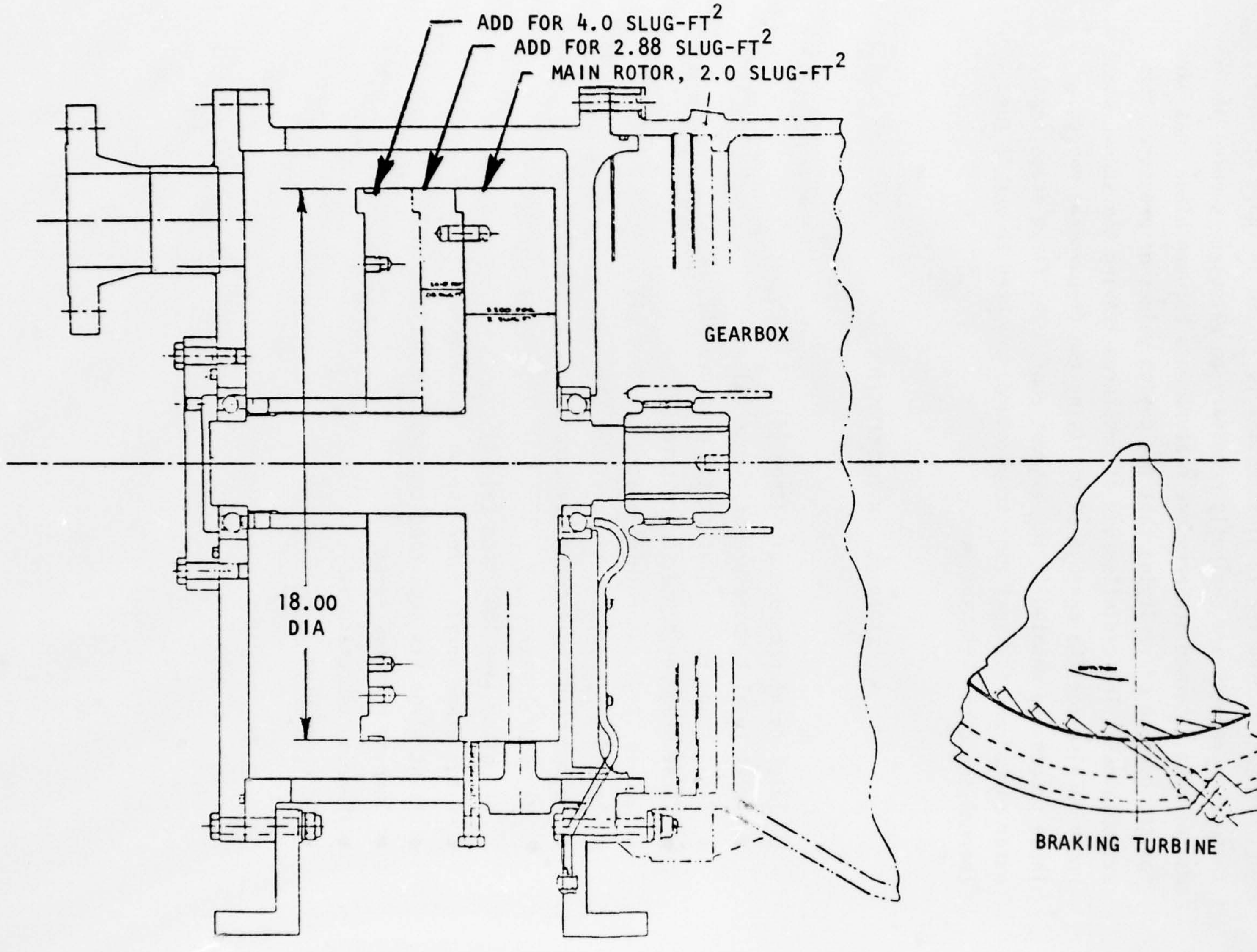


Figure 34. Inertia Simulator

4. INSTRUMENTATION

Instrumentation used for turbine testing is listed in Table XI. Most of these measurements are obviously necessary to calculate turbine performance. The intermediate pressure measurements between blade rows are for the purpose of verifying that the design values of pressure ratio are obtained. The radial survey of pressures leaving the second stage rotor is also for the purpose of verifying the free vortex design. These probes are mounted in the exhaust duct. The first stage nozzle outer shroud and tip seal ring temperatures are used to verify the thermal and cycle life analyses.

TABLE 11 INSTRUMENTATION

- Inlet total pressure at flange (1)
- Inlet total temperature of flange (1)
- First stage rotor inlet static pressure (1)
- First stage exit static pressure (1)
- Second stage rotor inlet static pressure (1)
- Second stage exit static pressure (1)
- Second stage exit total pressure and total temperature (4)
  a) radial survey (3)
  b) pitch line 180^{o} from (a) (1)
- First stage nozzle outer shroud temperature (1)
- First stage tip seal ring temperature (1)
- Turbine rotative speed
- Bearing temperatures
- Accelerometers

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